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AN INVESTIGATION OF RIGID FRAME BRIDGES
PART III

TESTS OF STRUCTURAL HINGES OF REINFORCED CONCRETE

A REPORT OF AN INVESTIGATION
CONDUCTED BY
THE ENGINEERING EXPERIMENT STATION
UNIVERSITY OF ILLINOIS
IN COOPERATION WITH
THE PORTLAND CEMENT ASSOCIATION

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AN INVESTIGATION OF RIGID FRAME BRIDGES

PART III

TESTS OF STRUCTURAL HINGES OF REINFORCED CONCRETE

I. Introduction

1. *Object and Scope of Investigation.*—In many cases it is highly desirable to introduce some form of articulation at the bases of the vertical members of concrete rigid frame bridges, particularly in long spans, where high deformation stresses would otherwise occur, due to temperature changes and shrinkage of the concrete. What constitutes a satisfactory hinge for such a structure can be determined only by laboratory tests and field experience. The purpose, therefore, of this investigation was to obtain information relative to the structural behavior of various types of hinges adaptable to concrete rigid frame bridges.

Previous tests of hinges for concrete structures have been limited to a type, similar to type 4 of this report, developed by the late Augustin Mesnager, a French engineer. Mesnager constructed two of his hinges and tested them to failure under vertical load combined with a forced rotation.* D. E. Parsons and A. H. Stang,† as well as G. C. Ernst,‡ made series of tests on the same type of hinge. Their work included axial and oblique load tests, as well as combined load and rotation tests, on specimens with and without concrete covering over the reinforcing bars at the center of rotation. A valuable discussion of the use of Mesnager hinges and the procedure to be followed in their design was presented in 1935 by B. Moreell.¶

A properly designed hinge, besides being sufficiently flexible to permit a given angular rotation, should be capable of withstanding compressive forces as well as shearing forces, be economical to construct, and require a minimum of maintenance. Seven types of hinges which it was thought might meet these requirements were selected for testing. The specimens were tested to determine their elastic and ultimate strength under vertical loads and under inclined loads such as occur at the hinges of a rigid frame bridge. Their

*"Experiences sur une semi-articulation pour routes en Beton Arme," M. Mesnager, Annales des Ponts et Chaussees, 1907.

†"Tests of Mesnager Hinges," D. E. Parsons and A. H. Stang, Journal of the American Concrete Institute, Jan.-Feb. 1935.

‡"Tests of Reinforced Concrete Hinges of the Mesnager Type," G. C. Ernst, Highway Research Board, Vol. 17, 1937.

¶"Articulations for Concrete Structures—the Mesnager Hinge," B. Moreell, Proc. A.C.I., Vol. 31, or Journal A.C.I., Vol. 6, pp. 368-381, 1935.

resistance to angular rotation was measured, and some were subjected to repeated rotation and then tested to failure under a vertical load combined with a forced rotation. Those specimens in which angular deformation occurred in the concrete during rotation of the hinge were tested to determine the effect of plastic flow on their resistance to rotation.

2. *Acknowledgments.*—The tests reported in this bulletin were a part of the investigation of rigid frame bridges conducted in the Arthur Newell Talbot Laboratory by the Engineering Experiment Station, University of Illinois, in cooperation with the Portland Cement Association. The work was under the administrative direction of Dean M. L. Enger, Director of the Station, Professor W. C. Huntington, Head of the Department of Civil Engineering, and W. M. Wilson, Research Professor of Structural Engineering. The Association was represented by an Advisory Committee consisting of Frank T. Sheets, President, and A. J. Boase, Manager of the Structural and Technical Bureau. The laboratory testing was under the general direction of F. E. Richart, Research Professor of Engineering Materials, to whom the author is indebted for helpful criticism and advice. The author is especially grateful to Professor Wilson for his careful examination of the manuscript and his suggestions for its improvement.

3. *Outline of Tests.*—Preliminary studies of rigid frame bridges indicated that the slope of the line of maximum thrust was in general about 1 vertical to 2 horizontal, and that an angular rotation as large as 0.004 radian could be expected at the bases of long span rigid frames freely hinged at these points. The tests were therefore planned on the basis of this information.

The testing program was divided into four series of tests, two series of load tests, a series of load and rotation tests, and a series of time-yield tests. All tests were made in duplicate. They are briefly described as follows:

Series V consisted of tests to failure under a vertical load passing through the axis of the hinge.

Series D consisted of oblique load tests to failure, in which the load was applied along a line inclined 26½ deg. to the vertical axis of the hinge.

Series R consisted of several different tests made on the same specimen. It included tests to determine the resistance of the hinge to an angular rotation of 0.004 radian, repeated rotation tests to the extent of 10 000 repetitions of angular movement, and, finally,

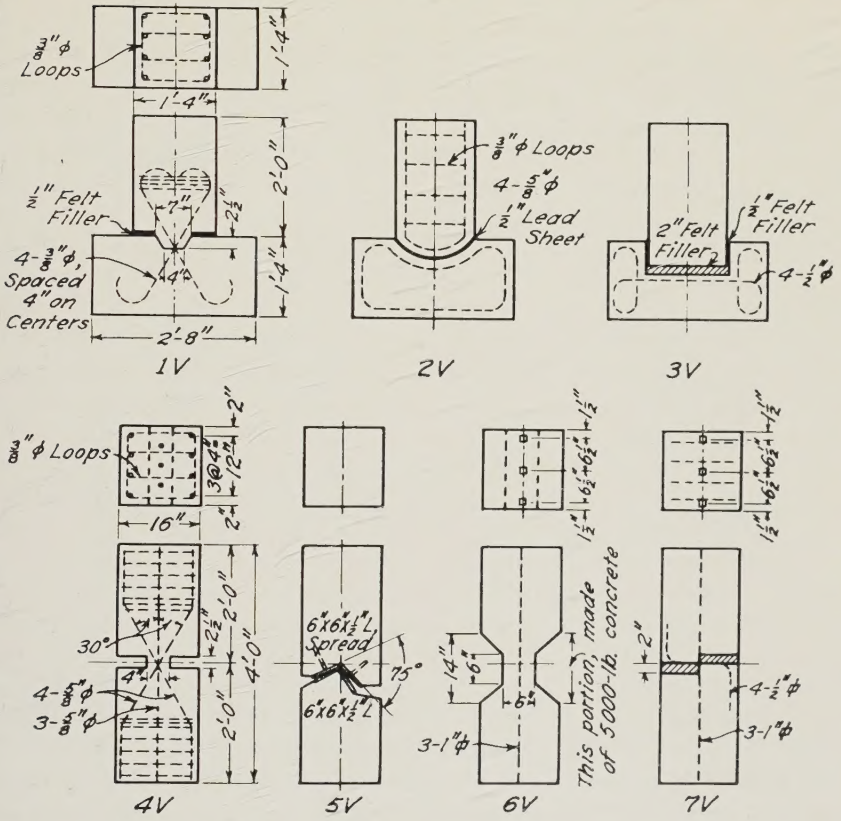

FIG. 1. VERTICAL LOAD SPECIMENS

tests to destruction, in which the hinge specimens were subjected to a vertical load and a forced rotation of 0.004 radian.

Series T consisted of time-yield tests to determine the effect of plastic yielding on the change in moment necessary to maintain a constant angular rotation over a period of time. Only those hinges in which the concrete participated in the action of the hinge were tested in this series.

II. DESCRIPTION OF SPECIMENS AND APPARATUS

4. *General Description of Test Specimens.*—The test specimens, and the details of the various types of hinges tested, are shown in Figs. 1, 2, and 3. The specimens shown in Fig. 1 were subjected to a vertical load passing through the axis of the hinge, those in Fig. 2 were subjected to an inclined load passing through the axis of hinge,

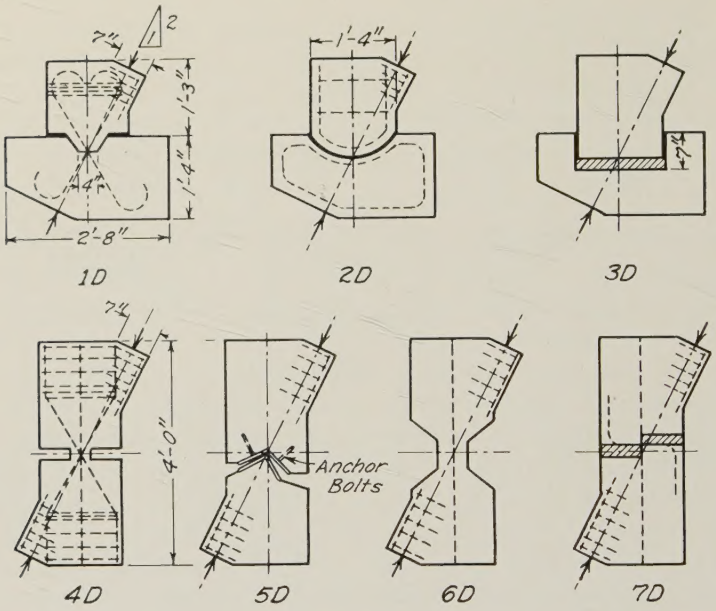

FIG. 2. OBLIQUE LOAD SPECIMENS

and those shown in Fig. 3 were subjected to a rotation. The various types of hinges are described briefly in the following paragraphs.

Type 1 is a conventional tongue-and-groove joint with ½ inch of impregnated felt filler placed between the vertical leg and the base on each side of the joint. Three-quarter-inch round deformed bars are crossed in pairs, and the pairs are spaced 4 inches apart.

Type 2 consists of a lead sheet between cylindrical surfaces of concrete.

Type 3 is another tongue-and-groove joint with a 2-inch impregnated felt filler underneath the vertical leg and ½ inch of felt along the sides.

Type 4 is a modified Mesnager hinge. It differs from the true Mesnager hinge in that it contains vertical bars in addition to the crossed bars. The bars are crossed in pairs and spaced 4 inches apart with a vertical bar between each crossed pair. This hinge was constructed and tested with and without concrete covering over the bars. The two variations were therefore designated as 4-c and 4-b, respectively.

Type 5 is a simple pivot consisting of two 6-in. x 6-in. x ½-in. structural steel angles anchored to the concrete, the corner of one

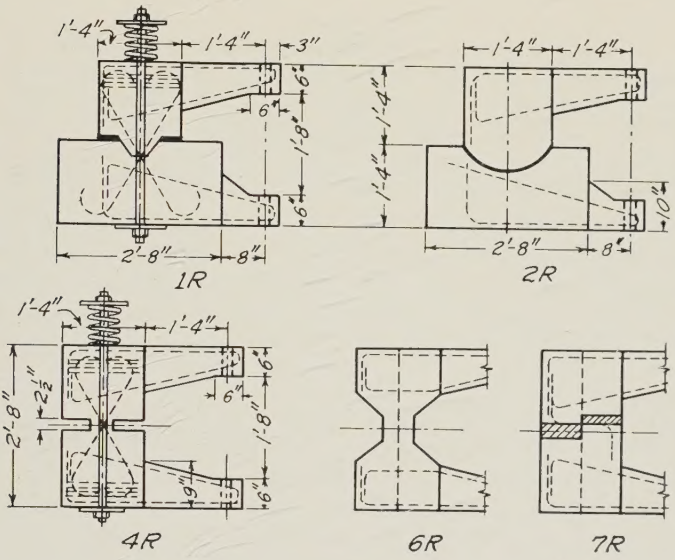

Fig. 3. Rotation and Time Load Specimens

bearing on the fillet of the other angle, which is spread slightly to provide clearance.

Type 6 represents a structural member having a deliberately reduced section of high strength concrete, reinforced at the center with 1-inch square vertical bars spaced 6½ inches apart.

Type 7 is a flexure fulcrum type of hinge in which 1-inch square vertical bars, spaced 6½ inches apart and protected by impregnated felt filler, are the flexure members, and ½-inch square horizontal bars resist shearing forces.

5. *Construction of Test Specimens.*—All of the test pieces were made from concrete designed to have a strength of 3500 lb. per sq. in. at 28 days, except the central portion of hinge 6, which was made from 5000 lb. concrete. The materials used were a standard portland cement, Wabash River torpedo sand, and gravel. The latter was graded from ¼ to 1 inch in particle size. The proportions of cement, sand, and gravel in the concrete used were 1:3¼:4, by weight, with a 1.1 water-cement ratio. This made a rather wet mix that was easily worked about the reinforcing bars. The average strength of the concrete in each specimen as determined from control cylinders is given in Table 2, page 11.

Steel forms were used except for some of the special details.

TABLE 1
AVERAGE PHYSICAL PROPERTIES OF REINFORCING STEEL

Specimen	Yield Point lb. per sq. in.	Ultimate Strength lb. per sq. in.	Elongation per cent	Reduction in Area per cent	Remarks
1-in. square	45 160	74 680	25.2	44.7	Deformed bar
¾-in. round	48 990	83 440	22.6	44.8	Deformed bar
⅝-in. round	44 530	76 830	21.4	55.0	Plain bar
½-in. square	47 120	75 380	21.7	44.0	Deformed bar
⅜-in. round	49 300	76 800	20.8	58.0	Plain bar

After 24 hours the forms were stripped, the specimens were wrapped in wet burlap and were cured in this moist condition for 7 days; they were then allowed to dry out in the laboratory air for a period of 21 days before testing. Three control cylinders were obtained (usually one from each batch) for each test specimen. They were cured in the same manner as the test specimen and tested dry when 28 days old.

The reinforcing steel was of intermediate grade, plain and deformed, having the physical properties listed in Table 1. Those hinges containing crossed bars were reinforced at and above the bends with ⅜-inch bars in the form of rectangular loops closely spaced and securely wired to the hinge steel.

Several of the hinges, types 1, 2, 3 and 5, because of the nature of their construction were poured in two operations, the lower half being poured one day and the upper half the next day. Most of the hinges involved no unusual construction difficulties; some difficulty, however, was experienced in constructing the type 5 hinges, and this perhaps warrants special comment on the method employed. In order to construct type 5 in its normal vertical position the lower half of the specimen was formed to the proper shape to support the pivot angle in its correct position; then, after the concrete had set up, the angle was grouted in place with a neat cement mortar. The bearing angle in the upper half was then supported on the pivot angle and held in position with wedges and form stops while the upper half was poured.

6. *Description of Rotation Apparatus.*—The rotation machine used for the service tests of hinges is shown in Fig. 4. It contained an adjustable eccentric cam operated by a variable-speed motor. One of the lever arms of the hinge specimen was clamped rigidly to the

TABLE 2
SUMMARY OF LOAD TESTS

Specimen	Vertical Axial Load lb.		Oblique Load lb.		Ratio	Load-Rotation lb.		Ratio
Type No.	Ultimate Load	Cylinder Strength	Ultimate Load	Cylinder Strength	Obliq. Ult. / Vert. Ult.	Ultimate Load	Cylinder Strength	L-R Ult. / Vert. Ult.
1 — 1	598 000	3250	443 000	4080	0.76	589 000	3505	0.89
2	653 000	3810	515 000	4330		530 000	4490	
Average	625 500	3530	479 000	4205		559 500	4000	
2 — 1	778 000	3330	416 000	3560	0.46	646 000	4353	0.75
2	950 000	3790	390 000	3530		645 000	4400	
Average	864 000	3560	403 000	3545		645 500	4375	
4-c— 1*	366 000	3660	220 000	4040	0.53	382 000	4570	0.90
2	423 000	4310	200 000	4100		326 000	3720	
Average	394 500	3985	210 000	4070		354 000	4145	
4-b— 1*	155 400	3950	119 000	4340	0.71	134 000	4140	0.82
2	173 000	4360	115 000	4420		135 000	4240	
Average	164 200	4155	117 000	4380		134 500	4190	
5 — 1	260 000	3700	180 000	3960	0.68	(†)		
2	291 000	4760	195 000	3800		(†)		
Average	275 500	4230	187 500	3880				
6 — 1	585 000	3950–5150	220 000	3670–4500	0.39	545 000	3470–5280	1.05
2	500 000	3310–5080	207 000	3780–4840		594 000	–5455	
Average	542 500	3630–5115	213 500	3725–4670		570 500	3470–5365	
7 — 1	224 000	4050	91 500	4440	0.39	250 000	4365	1.05
2	237 500	3800	90 000	4110		233 000	4385	
Average	230 750	3925	90 750	4275		241 500	4375	

*c indicates bars covered with concrete, b indicates bars bare.
†No tests for this method of loading.

base of the machine, while the other end was supported on 2-inch rollers and connected by means of a 1-inch bolt passing through a pair of calibrated car springs, one on each side of the lever arm, to a horizontal motion bar which was operated by the eccentric cam. The car springs were adjusted so that they were always in compression and operated in constant opposition to each other. This arrangement permitted a gradual angular movement of the hinge from 0 to a maximum of 0.004 radian in one direction to 0.004 radian in the other direction, and also provided a method of measuring the force required to rotate the hinge.

III. Description of Tests

7. *Vertical Load Tests.*—The hinge specimens were tested in a Southwark-Emery 3 000 000-lb. testing machine. Load was applied through a spherical block, fastened to the head of the testing machine,

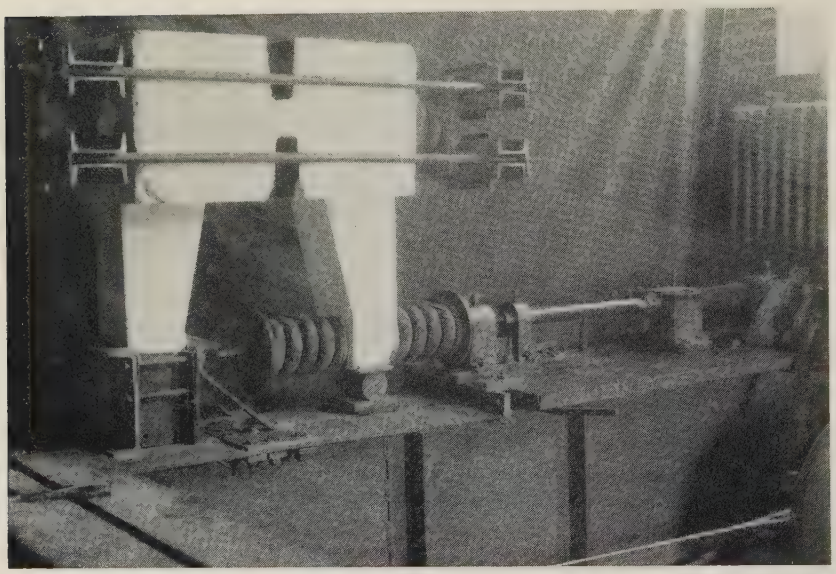

FIG. 4. VIEW OF REPEATED ROTATION APPARATUS WITH HINGE SPECIMEN IN PLACE

as shown in Fig. 5. A 1¼-inch machined plate was used to cap the specimen. Deformation of the entire length of the specimen was measured by four compressometers, one at each corner, held in place by steel bars clamped to the base and top of the specimen. Steel strains across the center of the hinge were measured on reinforcing bars that were close to the surface of the concrete, by means of a 10-inch Berry strain gage.

In order to center the specimen in the machine a small load was applied and the compression at each corner of the specimen was noted. If the deformation was not uniform the specimen was shifted until the eccentricity was eliminated, and then the spherical loading block was wedged in position.

Load was applied in increments of 20 000 and 25 000 lb. and a complete set of readings was obtained after each increment. The load was increased until failure seemed imminent, then all instruments were removed, and loading continued until complete failure of the specimen occurred.

8. *Oblique Load Tests.*—The oblique-load tests were conducted in much the same manner as the vertical-load tests, except that the specimen was inclined so that load was applied through the loading

FIG. 5. VERTICAL LOAD SPECIMEN IN TESTING MACHINE

shelves, as shown in Fig. 6. Deformations were measured directly across the hinge between the points *ab* and *a'b'*, indicated in Fig. 6, and diagonally across the hinge between points *ab'* and *a'b* to obtain a measure of the relative lateral movement of the upper and lower halves of the specimen. Steel strains were obtained in the same way as in the vertical-load tests. The specimens were centered in the testing machine by noting the compression at each corner, as indicated by the four extensometers shown in Fig. 6, and adjustments were made when necessary to eliminate any eccentricity.

9. *Rotation Tests.*—In this series of tests the resistance of the hinges to angular rotation was first measured; they were then given a service test consisting of about 10 000 cycles of reversed rotation through an angle of ± 0.004 radian, and finally they were tested to failure under a combined vertical load and a rotation of 0.004 radian.

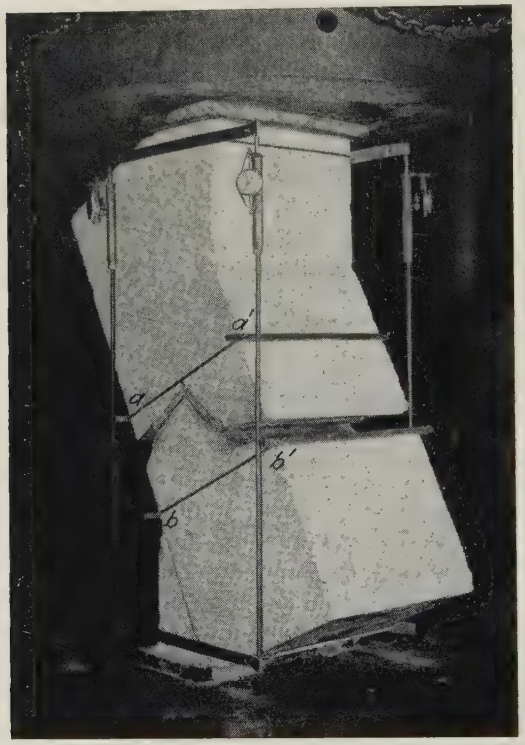

FIG. 6. OBLIQUE LOAD SPECIMEN IN TESTING MACHINE

A specimen was first loaded axially with calibrated car springs, the load being applied through steel bearing plates by means of an adjustable yoke, as shown in Fig. 7. Considerable care was exercised in applying the load so that it was absolutely central. Its position was carefully checked by the deformation dials at each corner of the specimen. The magnitude of this load was about one-third of the ultimate strength of the hinge, loaded in this manner, as estimated from the vertical load tests on similar specimens. This was considered a normal working load. The hinge was actuated by pulling the lever arms of the specimen together with a long 1-inch bolt passing through the lever arms and a calibrated spring. Compression of the spring was measured for various degrees of rotation until a maximum of 0.004 radian was produced. Knowing the force required to produce a given rotation, and the length of the lever arm, the moment resistance of the hinge was calculated. For hinges of type 2 this test was a matter of determining the moment necessary

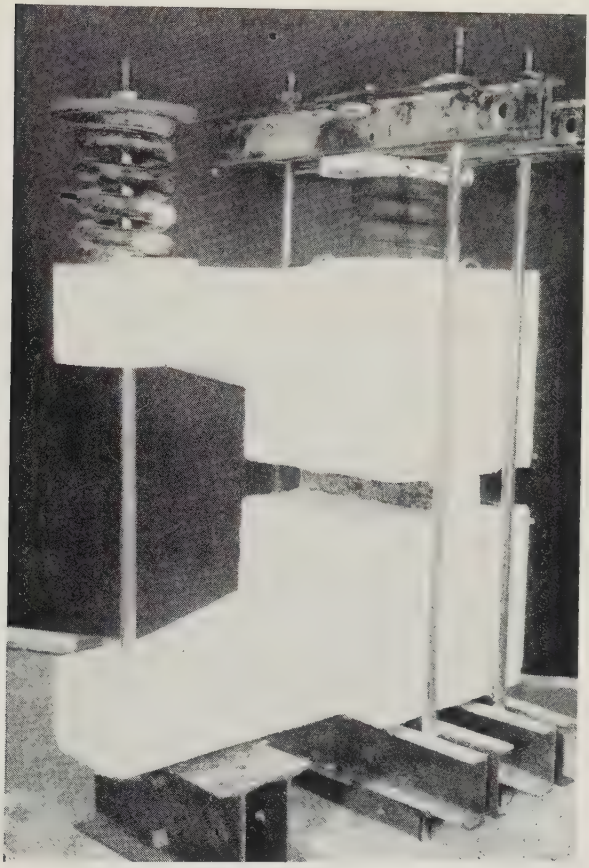

FIG. 7. ROTATION SPECIMEN EQUIPPED FOR TESTING

to overcome static friction between the lead and the concrete surfaces, and, since friction depends upon the pressure between the surfaces, the test was repeated for increasing axial loads.

The hinge specimen was next mounted in the rotation machine described in Section 6 and shown in Fig. 4, and the eccentric adjusted to produce an angular movement of the hinge in both directions from its normal position of 0.004 radian. The speed of the machine was regulated so as to produce about fifteen reversals of rotation per minute. During this test the hinge was carefully observed to detect any possible structural damage or decrease in resistance to rotation.

After the specimen had been subjected to 10 000 cycles it was removed from the rotation machine, placed in the 3 000 000-lb.

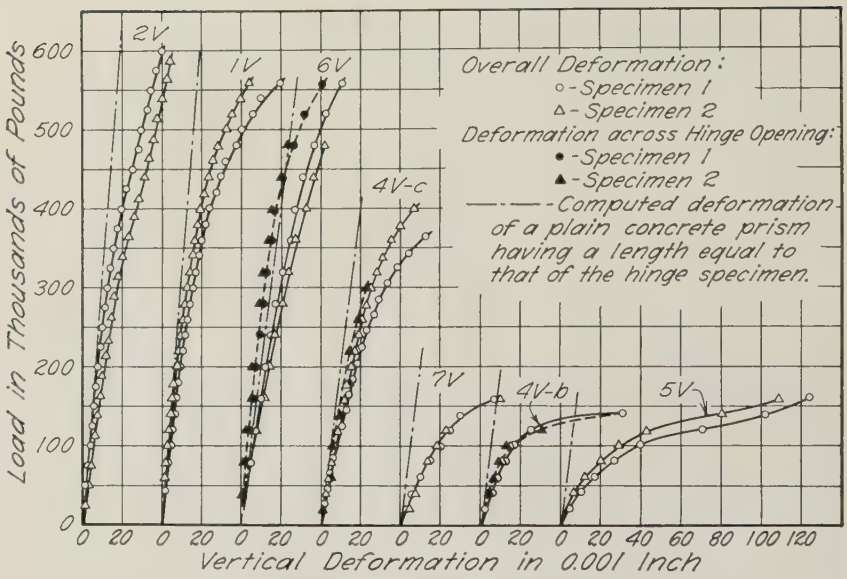

FIG. 8. LOAD–DEFORMATION CURVES FOR HINGES TESTED UNDER VERTICAL AXIAL LOADS

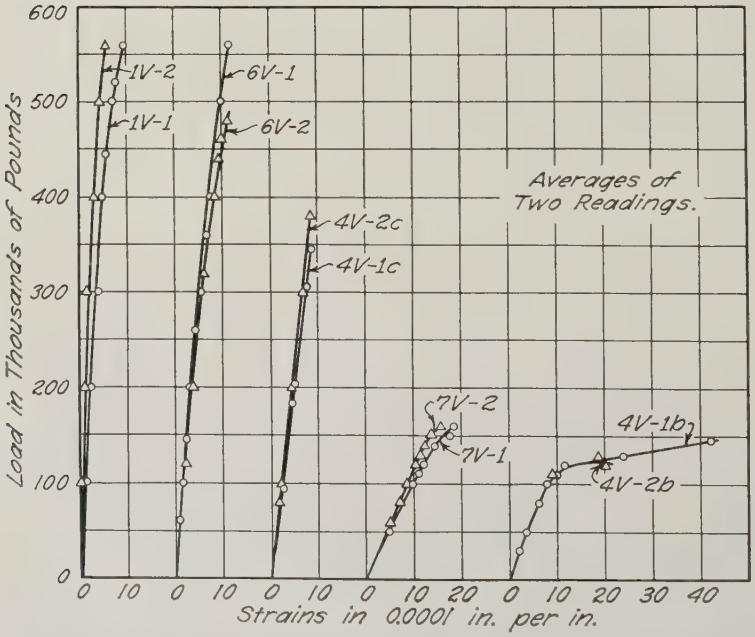

FIG. 9. LOAD–STRAIN DIAGRAMS FOR STEEL IN VERTICAL LOAD SPECIMENS

testing machine, and tested to failure under a static load. An initial rotation of 0.004 radian was induced at the beginning of this test by applying the load through tapered plates, one above and the other below the specimen. Enough load was applied to cause a rotation of 0.004 radian, the spherical blocks were then wedged in position to prevent further rotation, and the load increased until failure occurred.

10. *Time Yield Tests.*—Tests to determine the reduction in resisting moment of the hinge due to plastic yielding of the concrete for an angular rotation of 0.004 radian were performed on hinge types 4-c and 6. These, except for type 1, were the only hinges in which concrete directly participated in the action of the hinge. Type 1 was not tested because of its extreme rigidity, and because of the resulting damage to the specimen in rotating it. The testing apparatus and the method of rotating the hinge was the same as for the tests to determine the resistance of the hinge to angular rotation described in the preceding section. Each hinge was subjected to a sustained axial load corresponding to one-third of its estimated ultimate strength. The general testing procedure after the initial rotation of the hinge was to maintain the rotation constant by releasing the spring compression at the ends of the levers at frequent intervals during the test, and to measure the spring deformation immediately after each adjustment. The force exerted by the spring was determined from its calibration, and the resisting moment of the hinge was thus calculated.

IV. Results of Tests

11. *Strength of Hinges Subjected to Vertical Loads.*—Load–deformation curves for the vertical-load tests are shown in Fig. 8. The values plotted represent the average deformation as measured at the four corners of the specimen over its entire length, and in several instances the average deformation measured directly across the hinge opening. The curves are compared with the computed total deformation, shown by dot and dash lines, of a block of concrete with the same dimensions as those for the hinge specimens, and having an assumed modulus of elasticity of 3 500 000-lb. per sq. in. This comparison illustrates the relative stiffness of the various types of hinges when subjected to direct thrust. It is to be expected that a large percentage of the total deformation will occur across the hinge opening in the more flexible types of hinges. The data indicate that

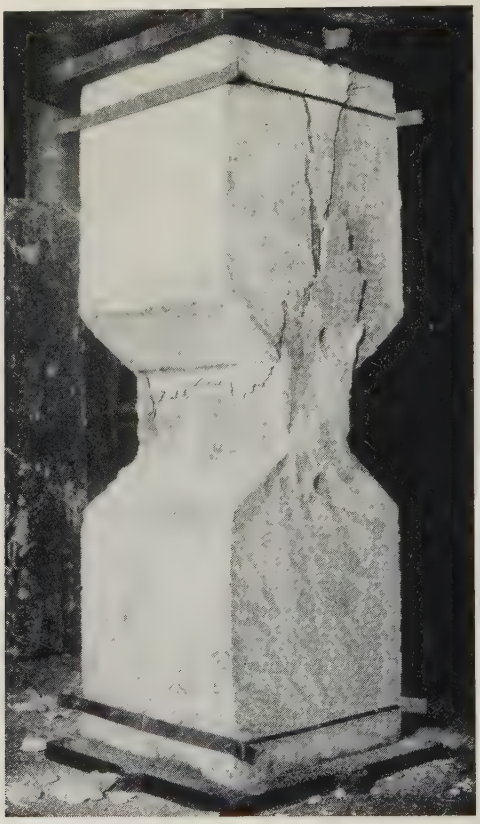

FIG. 10 (a) HINGE SPECIMEN AFTER TEST TO FAILURE

TABLE 3
COMPARISON OF COMPUTED MAXIMUM LOADS WITH TEST VALUES
Vertical Load Specimens

Hinge Type	Computed Maximum Load lb.	Test Value Maximum Load lb.	Ratio $\frac{\text{Test value}}{\text{Computed value}}$
1	515 700	625 500	1.21
2	912 000	864 000	0.95
4-c	390 700	394 500	1.01
4-b	135 700	164 200	1.21
5	541 000	275 500	0.51
6	492 000	542 500	1.10
7	135 480	230 750	1.70

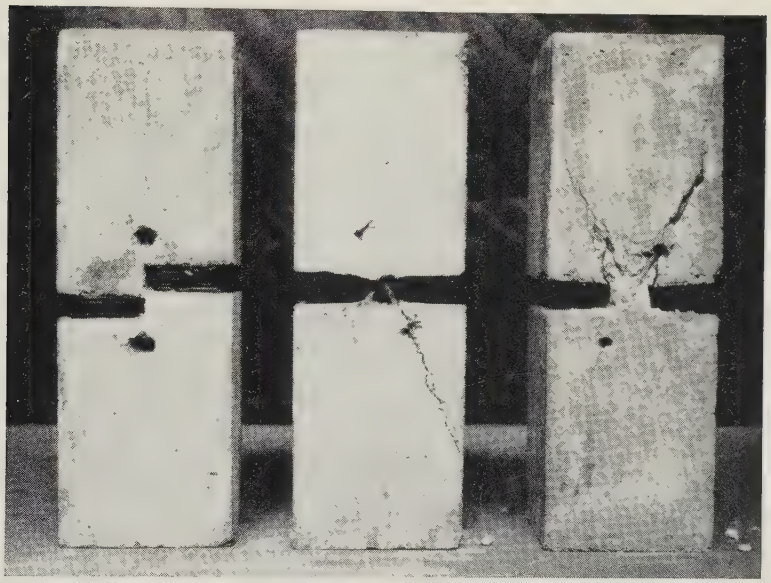

FIG. 10 (b) HINGE SPECIMENS AFTER TESTS TO FAILURE

practically all the deformation occurs across the hinge in types 4V-c within working loads, and in types 4V-b up to failure of the specimens. The results of specimen 3V are not included because it failed to sustain any appreciable load until practically all the filler underneath and along the sides was forced out. Further tests of this type of hinge were therefore discontinued. Load–strain curves for the steel bars crossing the hinge in types 1, 4, 6, and 7 are shown in Fig. 9, and the ultimate loads as well as the average strength of the control cylinders for each specimen are listed in Table 2.

The maximum vertical load that each hinge should support was computed on the basis of the concrete strength indicated by the control cylinders, and the yield strength of the reinforcement as given in Table 1. Since the length of bare bars in types 4-b and 7 was quite short, it was assumed in the computations that the full yield point of the steel might be developed. This made it unnecessary to apply the rather complex equations that have been developed for Mesnager hinges.* The values thus computed are compared with the average test values for each type of hinge in Table 3. It is inter-

*See Parsons, D. E. and Stang, A. H. "Tests of Mesnager Hinges," Proc. A.C.I. Vol. 31, pp. 304-330, 1935.

esting to note the high strength developed by several of the hinges as compared with the computed values. Type 7, for example, resisted a load 70 per cent greater than the computed value. The actual strength of type 5 was considerably less than the computed value because the distribution of stress in the concrete adjacent to the angles was not according to the manner assumed. Furthermore, it is possible that transverse reinforcing just above the upper angle and just below the lower angle would have strengthened the hinge materially.

Views of several of the specimens after failure are shown in Fig. 10. The manner of failure of each type is briefly described in the following paragraphs. In each instance the companion test exhibited a similar type of failure.

Hinge IV. Failure of type IV specimens was due to splitting of the concrete along the diagonal steel bars. Splitting was first in evidence at a load of about 450 000 lb. when fine cracks were observed along the plane of the steel. The ultimate loads were 598 000 lb. and 653 000 lb. for IV-1 and IV-2, respectively.

Hinge 2V. Type 2V specimens failed by crushing of the concrete in the upper half of the hinge, developing approximately the same strength as the control cylinders. Diagonal cracks in the lower half radiating out from the cylindrical surface appeared just before failure. There was no apparent damage to the lead plate. The ultimate loads were 778 000 lb. and 950 000 lb. for 2V-1 and 2V-2, respectively.

Hinge 3V. Type 3V specimens failed to withstand loads greater than 10 000 lb. without causing serious extrusion of the filler underneath and along the sides of the joint. For this reason tests of this type of hinge were discontinued.

Hinge 4V-b. Primary failure of type 4V-b specimens was due to yielding of the steel as indicated by scaling of the exposed bars at a load of 100 000 lb. Final failure was caused by splitting of the concrete along the diagonal bars. There was no evidence of buckling of the exposed bars even at the capacity load. The ultimate loads were 155 400 lb. and 173 000 lb. for specimens 4V-b1 and 4V-b2, respectively, and a view of failure is shown in Fig. 10(b).

Hinge 4V-c. Initial failure of the type 4V-c specimens was due to cracking along the diagonal steel. Crushing of the concrete covering and buckling of the steel within the covering, as shown in Fig. 10(b), occurred at the ultimate loads of 366 000 lb. and 423 000 lb. for specimens 4V-c1 and 4V-c2, respectively.

Hinge 5V. Failure of the type 5V specimens was due to the crushing of the concrete below the fillet of the lower pivot angle, followed by diagonal splitting of the lower half of the specimen. In the companion test a crack appeared in the upper half of the specimen, starting at the corner of the upper angle and extending about 12 inches. This occurred at about 0.8 of the maximum load, but did not contribute to the final failure. Examination of the structural angles after failure revealed a slight flattening of the edge of the pivot angle and only slight indentation in the fillet of the upper bearing angle. The ultimate loads were 260 000 lb. and 291 000 lb. for specimens 5V-1 and 5V-2, respectively.

Hinge 6V. The type 6V specimens exhibited a rather peculiar type of failure in which the outstanding portion of the specimen above the reduced section literally sheared off, as shown in Fig. 10. Cracks originated at the reentrant angle at the end of the reduced section, extended vertically several inches, and, as the load was increased, veered off toward the plane of the vertical steel bars. The ultimate loads were 585 000 lb. and 500 000 lb. for specimens 6V-1 and 6V-2, respectively.

Hinge 7V. Failure of the type 7V specimens was caused by the vertical bars yielding, followed closely by splitting of the concrete along the steel, as shown in Fig. 10(b). The ultimate loads were 224 000 lb. and 237 000 lb. for specimens 7V-1 and 7V-2, respectively.

12. *Strength of Hinges Subjected to Oblique Loads.*—The results of the oblique-load tests are reported in Table 2 and are shown in Figs. 11 and 12. The curves in Fig. 11(a) show the relative lateral movement of the upper and lower halves of the specimen due to the shearing component of the applied load. They are plotted in terms of oblique load and lateral deformation. Hinges 1D and 2D showed relatively little lateral movement of the upper half of the hinge relative to the lower half as compared with the more flexible types, and are therefore not included. The deformation in the longitudinal direction of the specimen is shown in the curves of Fig. 11(b). Load-strain diagrams for the steel bars in types 1, 4, 6 and 7 are given in Fig. 12. The curves labeled "North" refer to the diagonal bars which were almost parallel to the direction of the applied load, due to the inclination of the hinge in the testing machine. Consequently they received practically the full effect of the load as an axial force, whereas the curves labeled "South" refer to the diagonal bars which resisted the shearing component of the applied load. The break in the curves for specimens 4D-1b and 4D-2b (south) clearly indicate

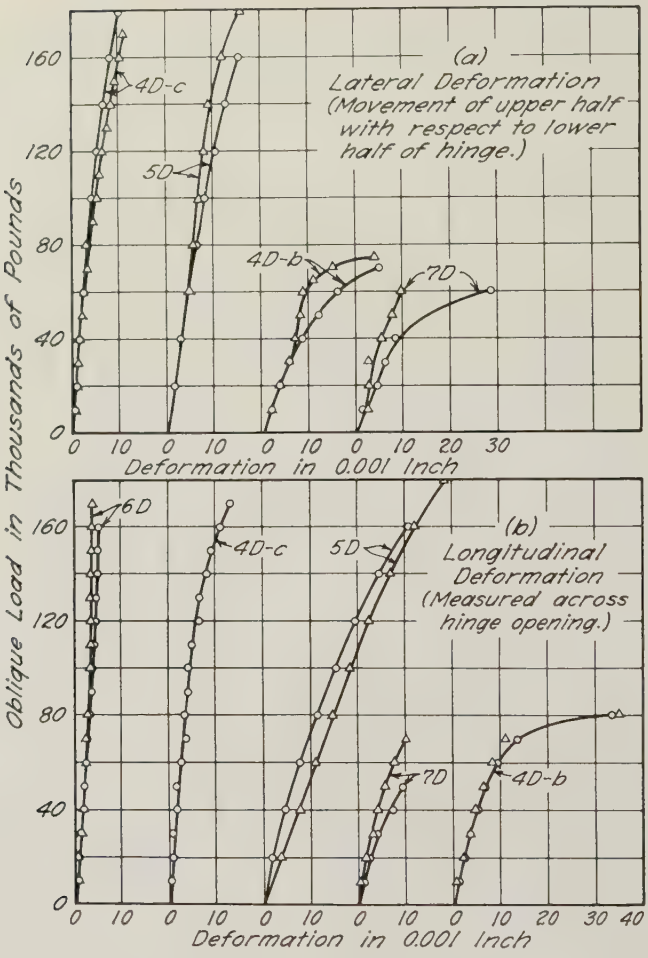

FIG. 11. LATERAL AND LONGITUDINAL DEFORMATIONS OF HINGES SUBJECTED TO OBLIQUE LOAD

bending of those bars at a load of 60 000 lb. These load-strain diagrams, together with those for specimens 4D-1c and 4D-2c, show very definitely the assistance of the concrete covering over the bars of type 4 in resisting an oblique thrust on the hinge.

The failure of the hinges subjected to an oblique load is due normally to a combination of a shearing thrust and a direct thrust, the individual effects of which are not readily separated. Failure in most of the hinges tested, however, was not so much influenced by shearing stresses as by other secondary effects, such as splitting

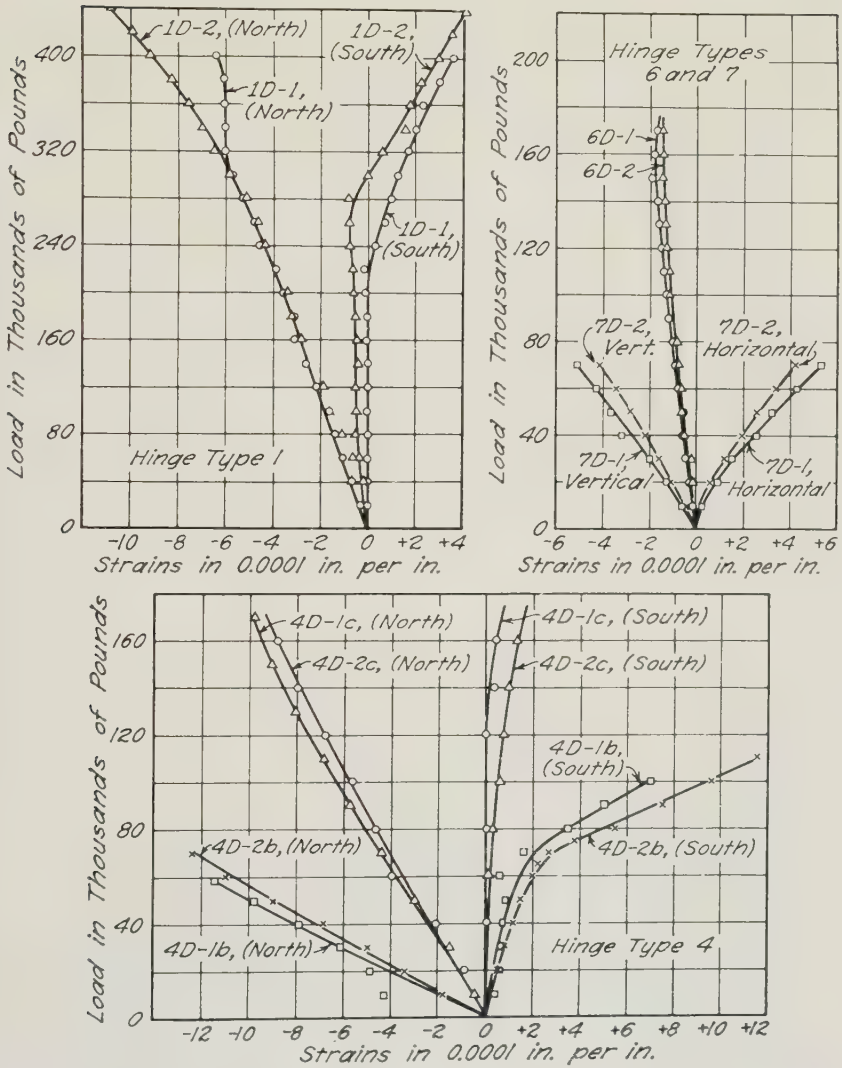

Fig. 12. Load–Strain Diagrams for Steel in Oblique Load Specimens

of the concrete, or buckling and bending of the steel bars; nevertheless it is of interest to resolve the maximum oblique load into its components and to observe the effect of a shearing force on the hinge by comparing the component parallel to the longitudinal axis of the specimen with the maximum strength of the hinge when tested under a vertical load. This has been done in Table 4.

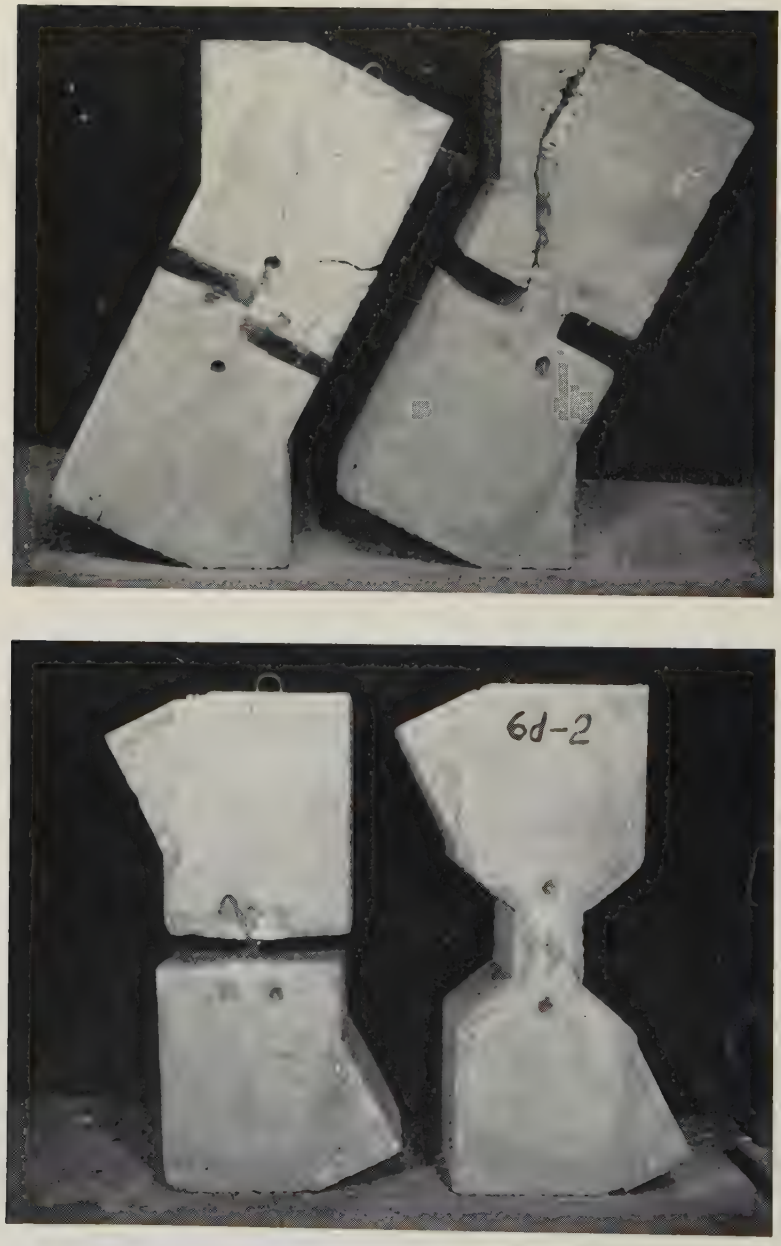

FIG. 13. HINGE SPECIMENS 7D, 4D-c, 4D-b AND 6D AFTER TESTS TO FAILURE

Table 4
Longitudinal and Lateral Components of Maximum Oblique Loads Compared with Maximum Vertical Loads
Maximum vertical loads in italics

Hinge Type	Maximum Oblique Loads lb.	Longitudinal Component lb.	Lateral Component lb.
1	479 000	428 000 *625 500*	214 000
2	403 100	360 000 *864 000*	180 000
4-c	210 000	188 000 *394 000*	94 000
4-b	117 000	104 800 *164 200*	52 400
5	187 500	168 000 *275 500*	84 000
6	213 500	191 000 *542 500*	95 500
7	90 750	81 200 *230 750*	40 600

Another comparison of the relative strengths of the hinges under oblique and vertical axial loads is shown as a ratio in Table 2.

It is interesting to note the relatively high value of the ratio exhibited by type 4-b as compared with type 4-c, which is a similar hinge except for the concrete covering over the bars at the hinge opening. The results might have been altered considerably had specimens of type 4-c been reinforced to prevent splitting of the concrete along the bars. The low value of the ratio shown by type 2 was due primarily to a premature failure of the base of each specimen, apparently because of its shape.

The manner of failure of several of the hinges is shown in Fig. 13, and a brief description of the failure of each specimen tested under an oblique load is given in the following paragraphs.

Hinge 1D. The specimens of type 1D failed by splitting of the concrete along the diagonal bars that were nearly parallel to the axis of the load in both the upper and lower portions of the hinge. Cracks first appeared at about ⅔ of the maximum load at the keyed joint. The ultimate loads were 443 000 lb. and 515 000 lb. for specimens 1D-1 and 1D-2, respectively.

Hinge 2D. The specimens of type 2D exhibited a general compression failure in the lower half of the specimen. There was no

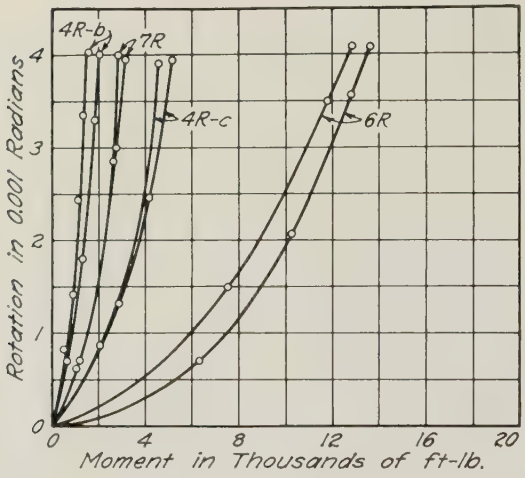

Fig. 14. Resistance of Hinges to Angular Rotation

apparent damage to the lead joint. The ultimate loads were 416 000 and 390 000 lb. for specimens 2D-1 and 2D-2, respectively.

Hinge 4D-b. Buckling and bending of the exposed bars, accompanied by splitting of the concrete along the diagonal bars, occurred in type 4D-b at ultimate loads of 119 000 lb. and 115 000 lb. for specimens 4D-b1 and 4D-b2, respectively.

Hinge 4D-c. Specimens of type 4D-c failed by splitting of the concrete along the diagonal bars that were in the direction of the load, as shown in Fig. 13. The concrete covering the bars at the hinge, however, was intact. The ultimate loads were 220 000 lb. and 200 000 lb. for specimens 4D-c1 and 4D-c2, respectively.

Hinge 5D. The upper half of the first specimen of type 5D split diagonally above the corner of the upper bearing angle, the angle apparently acting as a wedge to cause the separation. The lower half of the hinge was intact. The companion specimen failed by a combination of crushing and splitting below the pivot angle in the lower half of the specimen, the upper half being intact. The ultimate loads were 180 000 lb. and 190 000 lb. for specimens 5D-1 and 5D-2, respectively.

Hinge 6D. The specimens of type 6D exhibited a typical diagonal tension failure at the reduced section of the hinge, as shown in Fig. 13, at ultimate loads of 220 000 lb. and 207 000 lb. for specimens 6D-1 and 6D-2, respectively.

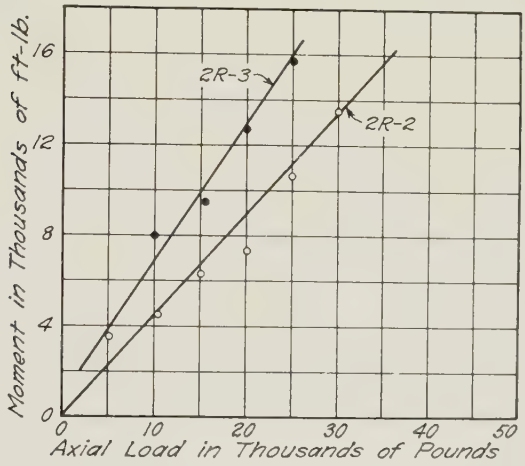

Fig. 15. Relation Between Axial Load and Resistance to Initial Rotation of Hinge Type 2

Hinge 7D. The failure of specimens of type 7D was caused by buckling of the vertical bars toward their unsupported side, that is, toward the side containing the felt filler, at the relatively low ultimate loads of 91 500 lb. and 90 000 lb. for specimens 7D-1 and 7D-2, respectively.

13. *Resistance of Hinges to Rotation.*—Moment–rotation curves plotted from data obtained from the tests described in Section 9 are given in Fig. 14 for types 4-b, 4-c, 6 and 7, and the relation between various axial loads and the moment required to overcome static friction in hinges of type 2 is shown in Fig. 15. In each case the data from tests of two specimens are plotted. Hinges of type 5 are not included, since they offer no resistance to rotation, type 3 was not tested, and type 1 was so rigid that the lever arms failed before any rotation was observed. In later tests of this hinge, in which rotation was forced by applying the load through tapered plates, it was necessary to exert a moment of approximately 25 000 ft. lb. in order to produce an angular movement of 0.004 radian. However, not until the base of the specimen was cracked and showed considerable evidence of splitting was there any appreciable rotation produced.

The friction characteristics of the two hinges of type 2 differ considerably, as might be expected, since it is difficult to reproduce

the same surface conditions from one specimen to the next. A vertical axial load of 20 000 lb. over a length of 16 in. is probably a normal working load for this type of hinge, and for such a load the hinge can be expected to develop a resisting moment of from 10 000 to 12 000 ft. lb. before the static friction between the cylindrical surfaces is overcome to permit any rotation. Resistance to rotation of the other types of hinges, excluding types 1 and 5, range from about 1800 ft. lb. for type 4-b, to about 12 500 ft. lb. for type 6 for an angular movement of 0.004 radian. Expressed in other words, hinges of type 6 offer about seven times more resistance to an angular rotation of 0.004 radian than the more flexible hinges of type 4-b. However, the degree of flexibility loses its importance when one considers that the moment of 12 500 ft. lb. developed by the stiffest of these hinges is small compared to the moment required to maintain fixity at the bases of a rigid frame bridge.

All of the hinges tested for repeated rotation, namely types 4-b, 4-c, 6 and 7, withstood 10 000 repetitions of a cycle in which the hinge was rotated from $+0.004$ radian to -0.004 radian under one-third of its static load capacity without evidence of progressive failure, and the moment required to produce this rotation was not excessive, neither did it change materially.

14. *Strength of Hinges Subjected to Combined Load and Rotation.*— The ultimate strength of hinges subjected to a combination of vertical load and rotation, in the manner described in Section 9, is given in Table 2, and views of the specimens after failure are shown in Fig. 16. From a comparison of these values with those listed under vertical-load tests, it would appear that an angular rotation of 0.004 radian does not greatly affect the ultimate strength of such hinges when subjected to vertical loads. Except for hinges of type 2, the decrease in strength that can be attributed to the effect of rotation amounts to but 15 per cent at the most; and in several instances there appears to be no decrease. Although specimens of type 1 were somewhat damaged structurally due to rotation, they nevertheless developed about 90 per cent of the strength of similar specimens tested under vertical load without rotation. Failure occurred in the base of the specimen, as shown in Fig. 16(b). The hinges of type 2 showed a marked reduction in ultimate strength after they were rotated. The concrete failed in compression with no apparent damage to the lead plate. The hinges of type 4-c showed a reduction in strength of about 10 per cent, if a correction is made for variation in the strength of the concrete as given by control cylinders. Failure of these

rotated specimens was very similar to the failure of those subjected to the vertical-load tests, namely, by crushing of the concrete covering and buckling of the steel within the covering. The same type of hinge, except with the bars exposed, showed a decrease in ultimate strength of 15 per cent due to the rotation. Failure was due to yielding of the hinge bars accompanied by splitting of the concrete along their embedded length, as shown in Fig. 16(a). Specimens of type 6R developed approximately the same ultimate strength as specimens of type 6V although failure was due to flexure as well as direct load, the specimen failing in compression at the reentrant angle adjacent to the reduced section of the hinge. The manner of failure is shown in Fig. 16(a). The hinges of type 7 also showed approximately the same ultimate strength for both vertical-load and load–rotation tests, and both showed similar types of failure, namely, by yielding of the vertical bars.

15. *Effect of Plastic Yielding on Resistance of Hinge to Rotation.*—The effect of plastic yielding of the concrete in hinges of types 4-c and 6 on their moment resistance for a rotation of 0.004 radian is shown in Fig. 17. The moment decreased rapidly during the first 24 to 48 hours, then decreased less rapidly and finally became practically constant after 30 days. The total reduction in moment amounted to about 85 per cent for type 4-c and about 63 per cent for type 6. The residual moment of type 4-c after a 30-day period of plastic yielding was about one-third of the normal resisting moment developed by the same type of hinge except with exposed bars at the hinge opening. The reduction in moment of type 6 made its resistance to a sustained rotation of 0.004 radian after a period of 30 days about the same as that of type 4-c for an initial rotation of the same magnitude.

16. *Discussion of Results.*—Tests were made on a number of types of hinges to determine their relative merits, with the result that some proved to be of good design whereas others were deficient in some respects. Moreover, the manner of failure of some hinges indicated how the design could be changed so as to increase the quality of the hinge without adding greatly to its cost. The addition of ties around the diagonal bars of hinge types 1, 4-b and 4-c, between the bend and the hinge opening, as pointed out by other investigators, would be desirable. The placing of shear steel in the form of spirals is also recommended for hinges of type 6. Hinges of type 5 are simple to construct and permit of free rotation. The manner of failure of these

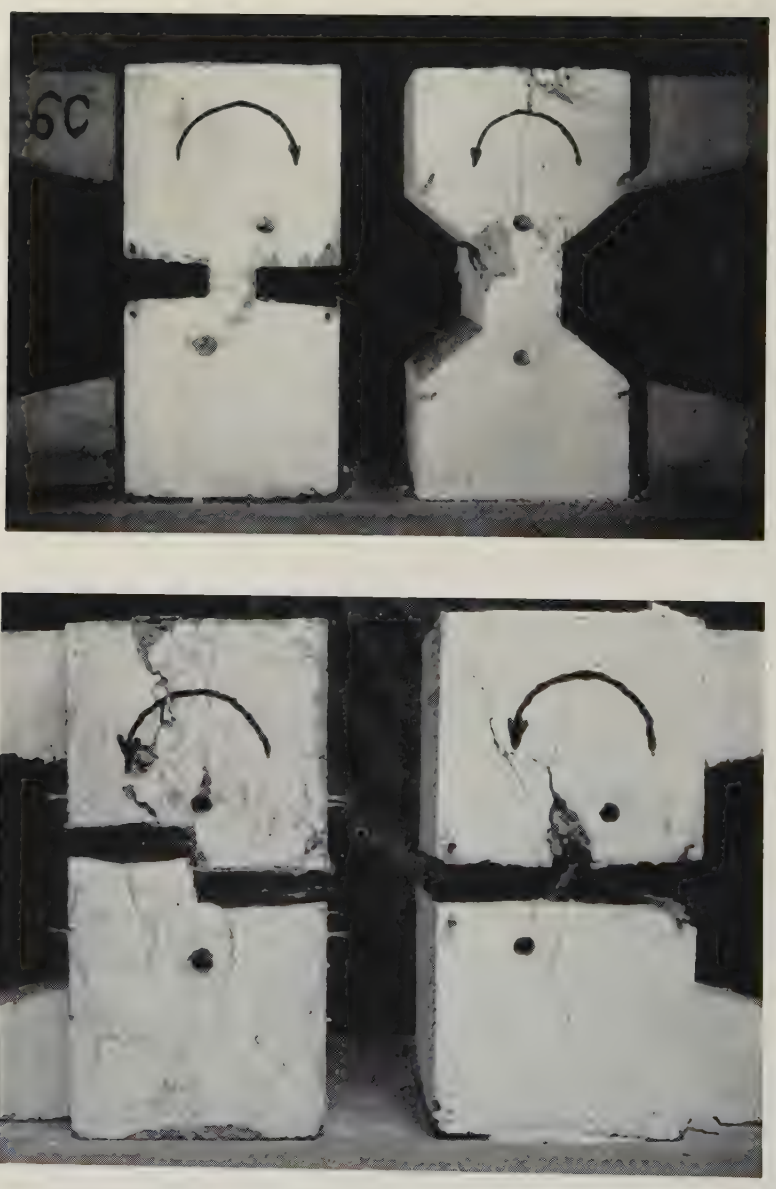

FIG. 16 (a) VIEWS OF HINGES AFTER LOAD–ROTATION TESTS TO FAILURE

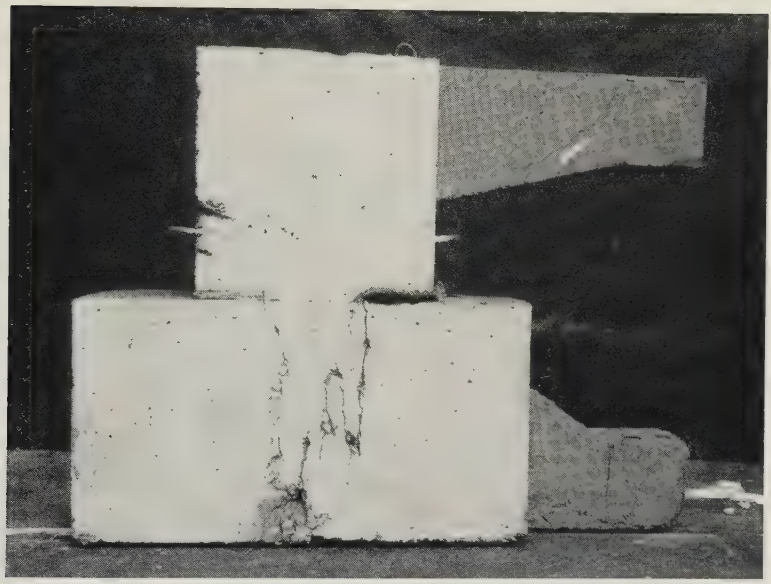

FIG. 16 (b) VIEW OF HINGE AFTER LOAD–ROTATION TEST TO FAILURE

hinges indicates that their strength would be increased by the use of transverse bars in back of the angles, or by other means of preventing the angle from splitting the concrete.

The hinges as designed developed strengths when tested under oblique loads that compare favorably with the strength required for the ordinary highway rigid frame bridge of medium spans. A study of rigid frame bridges designed for H-20 highway loading indicated probable maximum oblique thrusts at the hinged bases of these structures of 20 000 lb. to 60 000 lb. per foot of width for spans varying from 50 ft. to 100 ft. If a factor of safety of 3 is considered sufficient, all of the types tested, with the exception of 4-b and 7, would provide sufficient reserve strength under oblique loads to be considered adequate for spans up to possibly 70 ft., and with the suggested improvement in design they would be satisfactory for much greater spans. It should be noted that, except for hinges of type 4-b, the primary cause of failure of the hinge specimens tested under oblique loads was not over-stress in the steel bars, that is, yielding, so that a factor of safety is properly based on the ultimate strength.

The tests showed that the hinges had a greater resistance to vertical than to oblique loads. The thrust on the hinge of rigid

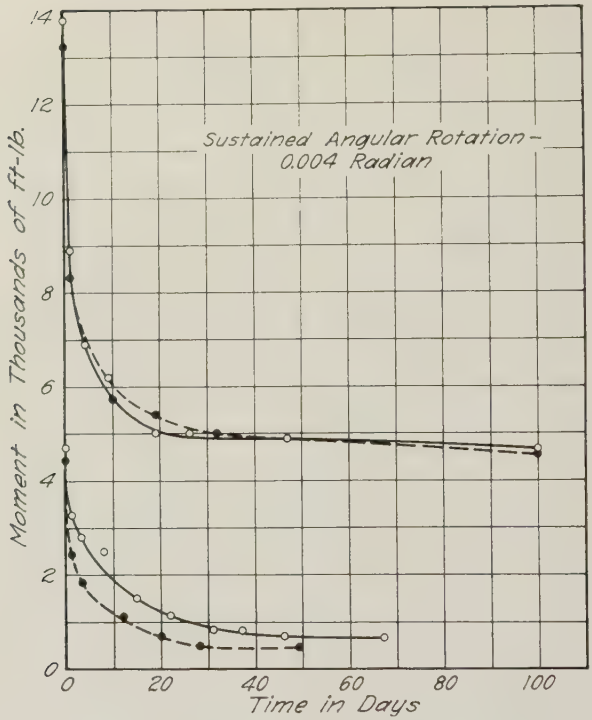

FIG. 17. REDUCTION OF RESISTING MOMENT DUE TO PLASTIC YIELDING—HINGE TYPES 4-C AND 6

frame bridges is oblique, but the angle between the two thrust lines, one for dead-load alone and the other for dead-load plus live-load, is not large. It would seem desirable, therefore, to decrease the angle between the thrust line and the axis of the greatest strength of the hinge by inclining the axis of the hinge in the actual structure. If this procedure were followed all of the hinges tested would have ample strength for the ordinary highway rigid frame bridge.

All of the hinges except type 1 were amply flexible to permit an angular rotation of as much as 0.004 radian. It is to be noted that the degree of flexibility of a hinge is not of so much importance as its ability to permit a given rotation without being structurally damaged. The flexural resistance of a hinge capable of a given angular rotation is in any case a small fraction of the flexural restraint necessary to prevent rotation at the bases of a rigid frame.

Probably about one-third of the total rotation at the hinges is due to changes in the length of the deck caused by shrinkage, the

effects of which occur over a relatively long period of time. For those hinges in which concrete participates in the action of the hinge, plastic flow provides a large measure of relief from this rotation as well as that due to dead load, so that generally these hinges need only provide for a momentary rotation of about one-half to two-thirds of the maximum anticipated rotation. For a highway bridge of 50-foot span this amounts to about 0.0015 radian.*

V. Summary and Conclusions

17. *Conclusions.*—Definite conclusions cannot readily be drawn from tests of a limited scope such as reported in this bulletin. The principal value of these studies is in providing a basis for further detailed experimental studies of the various types of hinges tested. The results of the tests do, however, point out the most desirable types of hinges for a particular rigid frame bridge, and they suggest certain improvements in design.

The following statements briefly summarize the important results of these tests.

(1) The measured strength of hinges tested under a vertical axial load was as great as and in some cases materially greater than the computed maximum strengths.

(2) Of the hinges tested those of types 1, 2, 4-c, 5 and 6 developed ultimate strengths when tested under an oblique load that compare favorably, on the basis of a safety factor of 3 or more, with the load requirements of the ordinary rigid frame highway bridge of medium single spans. With certain improvements in the hinge details, including larger bar sizes, increased strengths can be expected.

(3) All of the hinges, with the exception of type 1, were sufficiently flexible to permit an angular rotation as large as 0.004 radian without causing structural damage to the hinge.

(4) There was no evidence of deterioration of the hinges tested under repeated rotation after 10 000 reversals of 0.004 radian angular movement.

(5) The ultimate strength of the hinges was not materially affected by combining a forced rotation of 0.004 radian with a vertical load.

*See Bulletin No. 308, Part II, University of Illinois Engineering Experiment Station, Section 9, p. 37.

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Second Progress Report—Joint Investigation of Continuous Welded Rail

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Second Progress Report—Joint Investigation of Continuous Welded Rail

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I. Introduction

1. **Organization of Investigation.**—This investigation, begun in September 1937, is carried on under the joint auspices of the Association of American Railroads and the Engineering Experiment Station, University of Illinois. The Association of American Railroads, through the Committee on Rail of the American Railway Engineering Association, appointed the following advisory committee for the investigation:

J. C. Patterson, Chief Engineer Maintenance of Way, Erie Railroad (Chairman)
P. O. Ferris, Chief Engineer, Delaware & Hudson Railroad
G. M. Magee, Research Engineer, Engineering Division, Association of American Railroads
C. E. Morgan, Superintendent Work Equipment and Track Welding, Chicago, Milwaukee, St. Paul & Pacific Railroad
G. R. Smiley, Chief Engineer, Lousville & Nashville Railroad

At the University of Illinois the work was placed under the general direction of Professor H. F. Moore. Close contact is maintained with Professor A. N. Talbot and the Special Committee on Stresses in Railroad Track. One-quarter of the time of Professor H. R. Thomas, engineer of tests, and of Professor R. E. Cramer, metallurgist, is given to this investigation. The assistance of Messrs. S. W. Lyon, J. L. Bisesi, E. C. Bast, and N. J. Alleman is gratefully acknowledged. Acknowledgment is also made of the cooperation of the following representatives of welding companies in preparing specimens and offering constructive criticisms of test methods and the form of presentation of the results:

Lem Adams, Vice-President, Oxweld Railroad Service Company
C. A. Daley, Maintenance of Way Engineer, Air Reduction Sales Company
J. H. Deppeler, Chief Engineer, Metal & Thermit Corporation
H. C. Drake, Director of Research, Sperry Products, Inc.

2. **Previous Work of the Investigation.**—In the First Progress Report* of this investigation there were published data and preliminary results of (1) metallographic studies of welded joints in rails, (2) mechanical tests of specimens cut from rail metal, weld metal and metal in the junction zone between weld and rail, (3) tests of full-section welded-joint specimens under repeated wheel load and (4) drop tests and bend tests of full-section specimens of welded joints. The "rolling-load" testing machine for

* Proceedings of the American Railway Engineering Association, Vol. 40, pages 687–713 (1939).

subjecting rail-joint specimens to repeated wheel load was described, as was the testing rig for making bend tests of full-section rail joints. A bibliography on welded rails was printed as an appendix to the report. Attention was directed to the fact that all results obtained were preliminary and tentative.

3. **General Outline of the Work Since March 1939.**—In a general way the work reported in the First Progress Report has been continued. This is especially true in the case of rolling-load tests. Sixteen such tests have been completed, during which time 21,772,200 cycles of wheel load have been applied to specimens. This is equivalent to running the rolling-load machine continuously day and night for 252 days. A number of rolling-load tests of specimens with joint bars instead of welded joints have been tested. Further bend tests have been made and more micrographic study has been done. Plans have been made for the systematic reporting of welded rail failures in service.

4. **Types of Welded Joints Studied.**—Five different processes of welding rail have been studied. From all of these but one (Process D) joints with both 112-lb. rail and 131-lb. rail were furnished. Joints welded by Process D were furnished only with 131-lb. rails.

The detail of welding processes is covered rather fully in the First Progress Report. In general, the processes may be summed up as follows:

Process A. Rail joint welded by oxacetylene blow torch. A fusion weld made without pressure.

Process B. Similar to Process A but joints welded by a different company.

Process C. The Thermit process, using the intense heat generated by the transformation of aluminum to aluminum oxide. This process involves fusion, but after solidification of the molten metal has begun, pressure is applied.

Process D. An electric flash butt weld made with pressure. The weld is heated by striking an arc between the two rails to be joined and forcing the rails together at welding heat by pressure.

Process E. A gas butt weld with pressure. A special welding machine using oxacetylene gas is used and the rails are pressed together while welding.

In practically all the processes there is some heating for stress relieving or normalizing after the weld has been made and, of course, the grinding off of the excess metal over the tread and sides of the rail head.

II. Tests of Welded Joints Under Repeated Wheel Load

By N. J. Alleman and H. F. Moore

5. **Rolling-Load Testing Machine.**—The machine used for repeated-load tests of welded joints is called the "Rolling-load" machine. A halftone cut of the machine was shown in Fig. 8 of the First Progress Report. A diagrammatic sketch of the machine is shown in Fig. 1 of this second report for the benefit of those who may not have any other description available. The welded joint specimen S is pulled backward and forward under a wheel W, the load on which can be varied from zero to 75,000 lb. The load is applied through the lever L by means of a screw jack J, and is measured by the compression of the spring P. The maximum bending moment on the weld is equal to the load on the wheel times the distance from the center line of the weld to the point of contact of the wheel when the specimen is in its extreme left-hand position. A revolu-

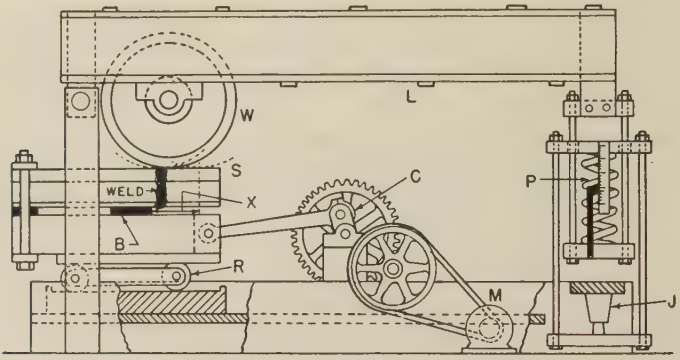


Fig. 1. Machine for Rolling Load Tests.

tion counter is attached to the crankshaft of the machine, and an automatic cut-off switch is operated by the drop of the lever *L* when a specimen breaks. The stroke of the machine is 12 in. and its speed is 60 strokes per minute.

6. **Procedure of Rolling-Load Tests.**—A number of specimens of each type of welded joint (not less than three) were tested for each of two weights of rail, 112-lb. and 131-lb.† The first specimen of each group was tested under a load of 75,000 lb. If the specimen broke before it had withstood 2,000,000 cycles of load a second specimen was put in under a lower load. If this specimen broke a third specimen was put in at a still smaller load and was tested to fracture, or until it had withstood 2,000,000 cycles of load. The selection of 2,000,000 cycles as the limiting number of cycles of stress is discussed in detail in the First Progress Report.

7. **Endurance Limit for Welded-Joint Specimens.**—The endurance limit for a given type of welded joint and given weight of rail is determined by drawing a stress-cycle (or S–N) diagram with wheel loads or bending moments as ordinates to an ordinary scale, and with number of cycles of stress for fracture as abscissas to a logarithmic scale. A line is drawn to fit the plotted points for specimens which break, and it is extended, if necessary, to intersect the vertical line for 2,000,000 cycles of load. The wheel load or bending moment represented by this intersection is taken as the endurance limit for 2,000,000 cycles of stress.

Figures 2 and 3 give S–N diagrams for the welds tested, and also S–N diagrams for rails without weld and rails with joint-bar joints. Whenever a specimen was removed from the machine without breaking, the plotted point for that specimen is marked with an arrow. Evidently an S–N diagram should not lie *below* such a plotted point. It is necessary in determining endurance limit for some of these S–N diagrams to estimate the endurance limit from rather unsatisfactory data. A few illustrations may be of service.

In the S–N diagram (Fig. 3) for gas fusion welds of 131-lb. rails the longest life of specimen was about 550,000 cycles. Three plotted points for the diagram fall closely on a straight line, and this line is extrapolated to 2,000,000 cycles. It is quite possible that the S–N diagram might flatten out like that in Fig. 2 for the gas pressure weld

† No specimens of the electric pressure butt-welded joints using 112-lb. rails have been supplied to the test party.

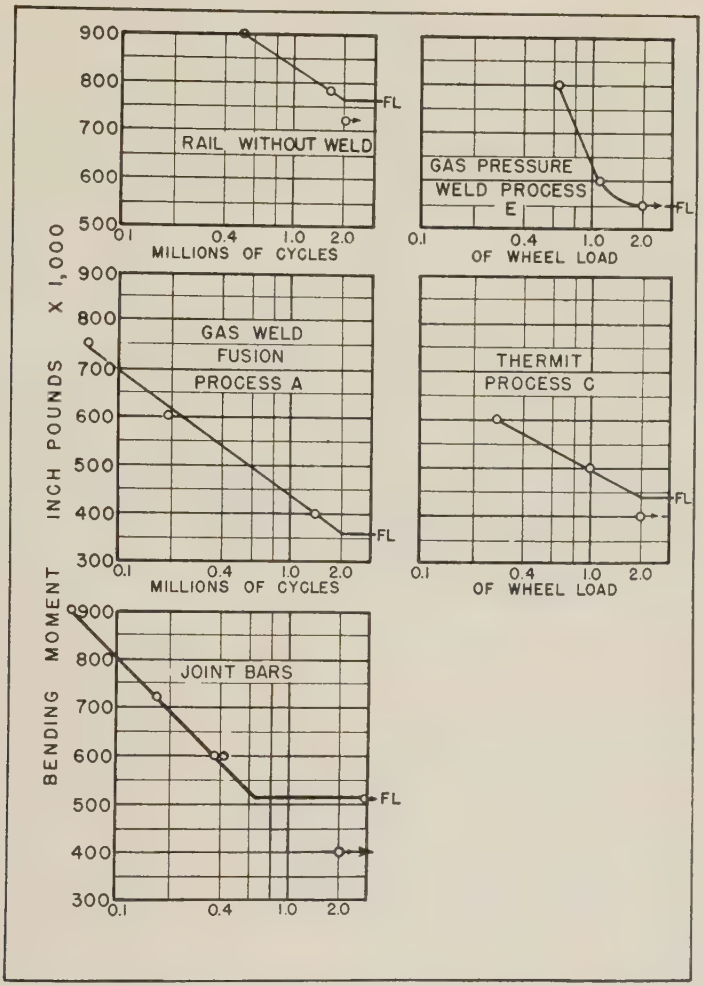


Fig. 2. Bending Moment-Cycle Diagrams for 112-lb. Rails, Joint Bars and Welded Rails. FL Denotes the Fatigue (Endurance) Limit.

specimens with 112-lb. rail. Then in Fig. 2 consider the S–N diagram for specimens with joint bars and 112-lb. rails. Evidently the S–N diagram should not pass below the plotted point for the specimen which did not fail after 3,028,600 cycles at 510,000 in.-lb., yet if the straight line connecting the three points for specimens which failed were extrapolated to 2,000,000 cycles, it would pass below the point for the unbroken specimen. The endurance limit is evidently less than 600,000 in.-lb. and not less than 510,000. It is estimated at 510,000.

8. **Types of Fracture of Welded Joints in Rolling-Load Tests.**—Figure 4 shows the three types of fractures which were found in welded joints, namely (a) failure by transverse fissure in head starting from some imperfection usually a minute

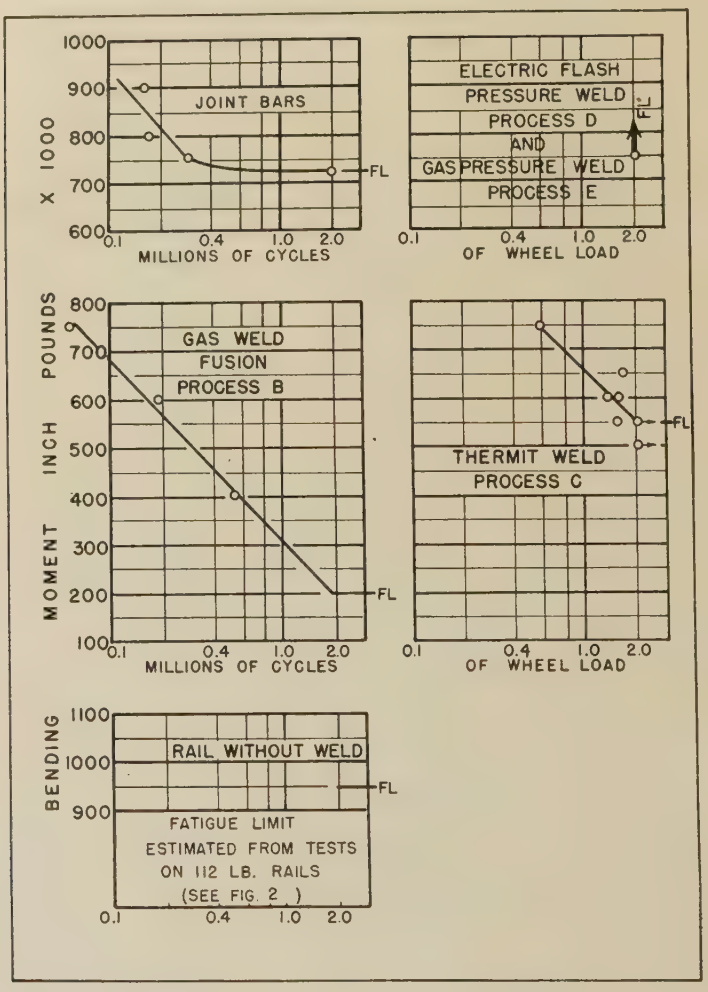


Fig. 3. Bending Moment-Cycle Diagrams for 131-lb. Rails, Joint Bars and Welded Rails. FL Denotes the Fatigue (Endurance) Limit.

crack, (b) failure starting in the web or at the junction of web and head—usually starting in some irregularity in welding surface which serves as a stress raiser, and (c) a fatigue failure by spreading crack starting near the corner of the tread of the rail—a conventional fatigue failure of the rail. Figure 4 (d) shows a typical rolling-load failure of a joint bar.

9. **Rolling-Load Tests of Joints Connected by Joint Bars.**—To secure a basis of comparison of the resistance of welded rail joints to repeated stress with that of joint-bar joints, a series of rolling-load tests on rails connected by joint bars is in progress. In this test the joint bars are carefully fitted and are tightened with a bolt tension of 20,000 to 25,000 lb. which is maintained as the rolling-load test proceeds.

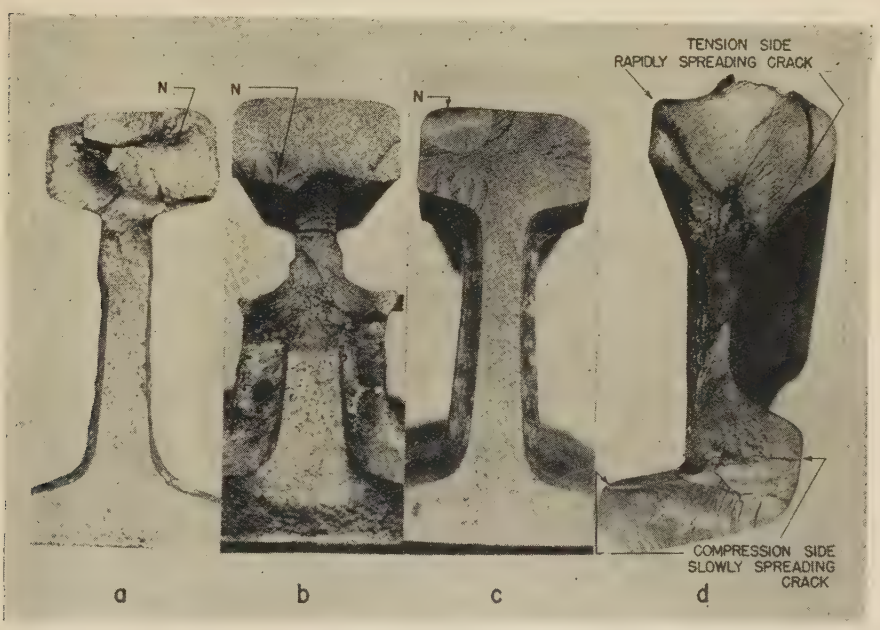


Fig. 4. Typical Fractures of Welded Joint Specimens in Rolling-Load Test.

a. Transverse fissure in head. Fissure started at N, and fracture occurred after 1,216,300 cycles of a 40,000-lb. wheel load.
b. Fissure started at N near junction of web and head, just outside the weld. Fractures occurred after 1,378,100 cycles of a 55,000-lb. wheel load.
c. "Fatigue" failure by spreading crack starting at N near corner of tread of rail. Crack started in coarse transverse grinding marks left on head of rail. Failure occurred after 650,000 cycles of a 75,000-lb. wheel load.
d. "Fatigue" failure of joint bar starting at upper and lower fishing surface after 56,100 cycles of a 75,000-lb. wheel load. Bending moment 900,000 in.-lb.

The specimen is so placed that the gap between the rails is at the edge of the block on which the rail rests and the path under the wheel load extends from this point to a point an inch or two from the free end of the specimen, which is mounted as a cantilever.

The rail joints with joint bars were placed in the machine the same way as the rail specimens. It is recognized that this throws tensile stress to the top of the joint bar and compressive stress to the bottom and that the cycle of the stress is quite different from that which has to be withstood by the joint bar in service. However, it was felt that the best comparison of strength under repeated load would be obtained by subjecting welded joints and joint-bar joints to the same testing conditions, namely, a cycle of bending moment varying from zero to maximum, with tension on the top side and compression on the bottom of the specimen.

The joint bars used in the test specimens were of heat-treated steel, and bars from the same lot were used for the 112-lb. joints. The joint bars for the 131-lb. rails were from two lots. It is recognized that there would be a wide difference between strength of different joint bars with different heat treatments and also that variations of bolt tension may produce wide differences in fatigue strength, as Professor Talbot's investigation has shown. These tests may be considered as representing test joints with a joint bar made of strong steel, carefully fitted to the rail and with a uniform bolt tension

carefully maintained. (Note: The test results for the two lots of 131-lb. joint bars fall on the same curve in Fig. 3). This would seem to represent good practice in joint bars. Perhaps it might be considered that the joints with joint bars were made up with care corresponding to the care with which the welded joints were made. In any event the values from the few tests which it has been possible to make thus far should be regarded as indicative of general ranges of figures rather than a basis for sharp comparisons of strengths.

10. **Fatigue Failures in Joint Bars.**—In the joint-bar joints the final failure was always by a fatigue crack on the tension side. In one or two cases it was observed that fatigue cracks *started* on the compression side but progressed very slowly. In some cases the fatigue cracks on the compression side appeared first, and were followed by rapidly spreading cracks on the tension side. This is in agreement with the cracking observed in some joint bars in service. However, as is commonly assumed today, a fatigue crack starts under repeated shearing stress and tends to spread most rapidly under tensile stress. In view of this the behavior of joint-bar test joints is quite natural. Shearing stress is very nearly as severe on the compression side of the bar as on the tensile side. There is no reason why a crack could not be expected to *start* under either compression or tension. However, the rate of spread of the crack is distinctly diminished by the closing up of the crack on the compressive side while on the tension side there is a tendency for it to open. (See Fig. 4 (d)).

11. **Results of Rolling-Load Tests.**—As has been previously noted the number of rolling-load tests which have been carried out is relatively small. The endurance limits which have been determined must be regarded as at best only roughly approximate. In a general way it may be said that in the rolling-load tests so far carried out, the electric butt pressure welds seemed to develop strength almost, if not quite, up to that of the rail itself; the gas pressure welds were but little, if any, inferior to the electric butt pressure welds; the Thermit weld specimens and the joint-bar specimens had almost the same strength; and the gas fusion welds tested showed lower strength than the joint-bar tests. Table 1 shows results of rolling-load tests.

12. **Bend Tests on Rail Joints.**—Table 2 gives the results of bend tests on welded-rail specimens made up to date. The testing rig for bend test is shown in detail in the First Progress Report and all the specimens were tested under a 64-in. span with two equal loads spaced 6 in. either side of the center of length of the specimen. Thus the middle 12 in. of length of the specimen was under uniform bending. In Table 2 the maximum load before failure would be an indication of the strength of the joint, while the energy for fracture is taken as a measure of its toughness. The deflection at failure might be taken as a measure of its static ductility. It will be noted that the electric pressure weld and the gas pressure weld actually developed a higher load than a minimum value for crack-free rails tested for the Rails Investigation. They also developed higher energy for fracture in 131-lb. specimens. It is noted that the joints with joint bars developed a high amount of energy for fracture.

13. **Bend Tests and Rolling-Load Tests as Criteria of Strength of Joint.** In the study of fatigue of metals it has been found by many investigators that for many methods there seems to be a correlation between tensile strength and fatigue strength. A study of the results of rolling-load tests and bend tests was made to see whether any such correlation can be traced in the joints tested. It was found that the data were not sufficient nor the results consistent enough to justify any precise quantitative correlation, but the relative order of results for the two tests is shown graphically

Table 1

Endurance Limits for 2,000,000 Cycles of Rolling Wheel Load for Welded Rail Specimens, Rail Specimens Without Welds and Specimen Joints Using Joint Bars

During test head of rail subjected to cycles of tensile stress varying from zero to a maximum

Type of Joint	At Endurance Limit				Rolling-load Strength Ratio*	Remarks
	Wheel Load, lb.	Maximum Moment, in.-lb.	Maximum Flexural Stress lb. per sq. in.	Maximum Shearing Stress lb. per sq. in.	percent	
Joints with 112-lb. Rails						
Rail without weld	63,000	756,000	42,000	25,600	100	Based on bending moment
Rail with joint bar	41,700	510,000	(1)	(1)	66	Based on bending moment
Gas fusion weld– Process A	36,000	360,000	20,000	11,700	51	Based on wheel load, failed by internal fissure
Thermit weld– Process C	44,000	440,000	24,300	14,400	58	Based on bending moment
Gas pressure weld—Process E	55,000	550,000	30,600	17,900	73	Based on bending moment
Joints with 131-lb. Rails						
Rail without weld	(2) 79,000	(2) 948,000	(2) 42,000	24,200	100	Based on bending moment
Rail with joint bar	60,400	725,000	(1)	(1)	76	Based on bending moment
Gas fusion weld—Process B	20,000	200,000	8,800	6,100	25	Based on wheel load; failed by internal fissure
Thermit weld—Process C	55,000	550,000	24,400	17,000	70	Based on shearing stress; Web failure.
Electric pressure weld—Process D	More than 75,000	More than 750,000	More than 33,100	More than 19,500	More than 77	Based on bending moment
Gas pressure weld—Process E	More than 75,000	More than 750,000	More than 33,100	More than 19,500	More than 77	Based on bending moment

* Ratio of (Endurance Limit for specimen) to (Endurance Limit for Rail without Weld).

(1) Flexural stress in joint bars not computed; joint bars and rails made of different strength stee

(2) Estimated on the assumption that 131-lb. rail will develop as high flexural stress at endurance limit as 112-lb. rail. Maximum load which can be applied to machine, 75,000 lb.

TABLE 2

RESULTS OF BEND TESTS OF WELDED RAIL SPECIMENS, RAIL SPECIMENS WITHOUT WELDS AND SPECIMEN JOINTS USING JOINT BARS

Specimens were tested head-down unless otherwise noted

Specimen		Maximum Load before Fracture, lb.	Energy for Fracture, ft.-lb.	Deflection at Fracture, in.	Bend-Strength Ratio* percent	Remarks
Number	Classification					
	Joints With 112-lb. Rails					
----------	Rail without weld**----------	225,000	75,000	4.0	100	
----------	Joint with joint bar----------	151,100	193,500	20.0	67	
----------	do. (tested head-up)----------	154,400	69,300	6.8	----------	No head-up tests of rail without weld for comparison
W 4	Gas fusion weld—Process A----------	124,200	4,300	0.7	55	
W113	Gas fusion weld—Process B----------	140,500	6,900	0.9	62	
W253	Thermit Weld—Process C----------	193,900	23,370	2.1	86	
W404						
W405	Gas pressure weld—Process E----------	228,400	57,800	3.9	101	Average of two tests
	Joints With 131-lb. Rails					
----------	Rail without weld**----------	300,000	100,000	5.0	100	
----------	Joint with joint bar----------	201,200	111,200	9.2	67	
----------	do. (tested head-up)----------	202,700	42,300	3.3	----------	No head-up tests of rail without weld for comparison.
W 22	Gas fusion weld—Process A----------	168,500	7,360	0.5	56	
W105	Gas fusion weld—Process B----------	79,400	867	0.2	26	
W203	Thermit weld—Process C Lot 1----------	231,500	22,200	1.65	77	
W223	Thermit weld—Process C Lot 2----------	254,000	33,800	2.2	85	Average of two tests.
W226						
W311	Electric pressure weld—Process D----------	315,000	121,500	6.0	105	
W451	Gas pressure weld—Process E----------	320,900	177,800	7.9	107	Average of two tests.
W452						

* Ratio of (Maximum load for specimen) to (Maximum load for rail without weld).
** Values for rails without welds taken from records of bend tests made by the Rails Investigation. Minimum values for crack-free rails recorded here.

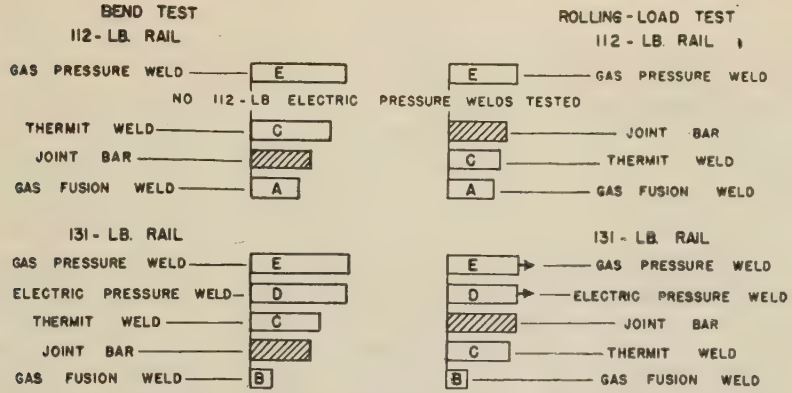


Fig. 5. Relative Strengths of Joints as Shown by Bend Test and Rolling-Load Test.
Letters on bars designate welding process.
Strength of Thermit weld joints in 131-lb. rails based on average for two lots of specimens.

in Fig. 5, in which the lengths of bars are proportional to the average strength ratios found. Attention is called to the fact that rolling-load tests for Processes D and E (131-lb. rail) merely indicated values greater than a certain amount. This is shown by arrows attached to the bars. An interesting fact in connection with this figure is that the bend tests and the rolling-load tests arranged the different types of weld in the same order of merit. It may be added here that taking both the bend tests and rolling-load tests into consideration it seems best to regard Processes D and E of so nearly equal value that a discrimination can not be made between them at present.

The correlation between order of merit as shown by the bend tests and that shown by the rolling-load tests is interesting. In view of the desirability of a reliable acceptance test for welded-rail joints further experimental study seems worth while. However, it would be unwise to conclude that such a correlation has been definitely established by these relatively few tests.

14. **Proposed Study of Welded Rail Failures in Service.**—A proposed questionnaire on the behavior of welded rails in service has been submitted to the Advisory Committee on Continuous Welded Rail with the request that some action may be taken so that this questionnaire or some other form may be sent to the railroads having welded rail in track. It would afford a source of systematic information on welded joint behavior in service, covering such items as first cost, maintenance cost, riding qualities, wear, and failures. At best, the laboratory tests of the strength of continuous welded rails indicate the order of magnitude of strength which may be expected as compared with the strength of rails without weld or with the strength of joint-bar joints in rails. Variations due to service conditions and variations in quality of the weld in joints made would have to be studied through service tests.

III. Metallographic Tests

By R. E. Cramer, assisted by E. C. Bast

15. **Second Lot of Thermit Welded Joints.**—During the past year metallographic studies have been made on a group of Thermit rail joints welded in May 1939. These welds differed from previous Thermit welds studied in two respects. First, the

molds had been changed to prevent the trapping of slag and loose mold sand at the junction of the molten weld metal and the rail steel near the lower portion of the rail head. Second, these welds were not post heated or given the strain relief heating to 1,100 deg. F. as were the previous Thermit welds.

16. **Hardness Tests, Etch Tests and Metallographic Examination.**—The rail heads in the test joints furnished were prepared for Rockwell "C" hardness tests over three areas: (1) on the side of the head, (2) on a horizontal surface 3/16 in. below the tread of the rail, and (3) on a horizontal surface 3/16 in. above the junc-

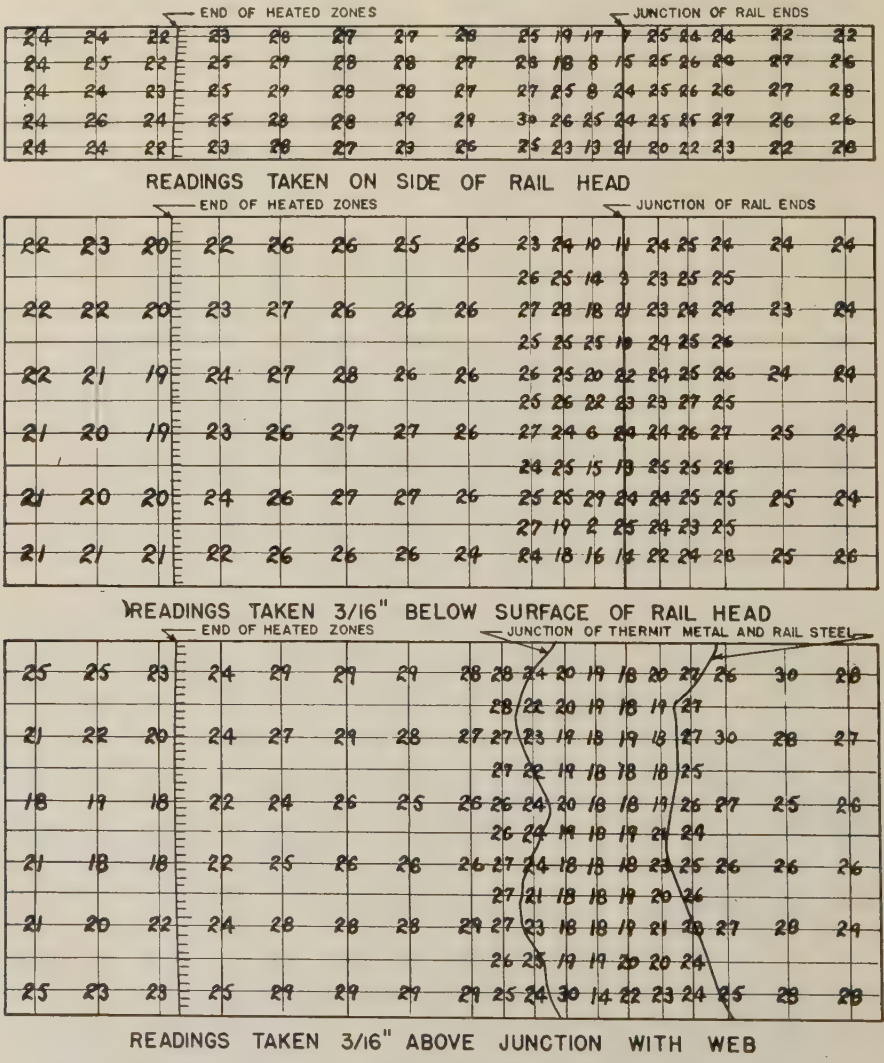

END OF HEATED ZONES
JUNCTION OF RAIL ENDS
READINGS TAKEN ON SIDE OF RAIL HEAD
END OF HEATED ZONES
JUNCTION OF RAIL ENDS
READINGS TAKEN 3/16" BELOW SURFACE OF RAIL HEAD
END OF HEATED ZONES
JUNCTION OF THERMIT METAL AND RAIL STEEL
READINGS TAKEN 3/16" ABOVE JUNCTION WITH WEB

Fig. 6. Hardness Tests on Welded Rail Joint Made by Metal and Thermit Corporation New Type Weld No. W220. 131-lb. Section. Rockwell C Scale.

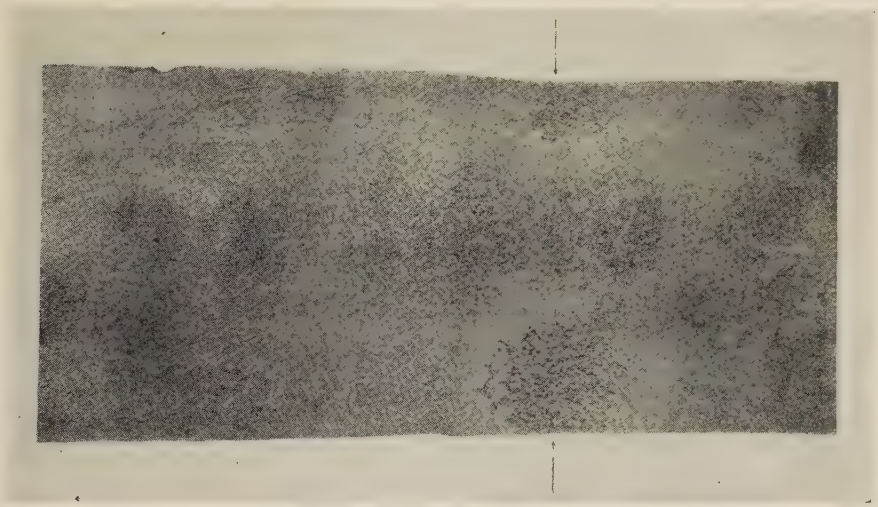


Fig. 7. Etched Horizontal Section of Weld ⅛ Inch Below the Tread.

Three-fourths natural size. Etched with hot Hydrocholoric Acid. The acid has brought out marks at both sides of the head at the junction of the rail ends. These marks are caused by oxide at the grain boundaries, as shown in Fig. 8.

Fig. 8. Photomicrograph of Metal Showing Oxide Inclusions at Grain Boundaries. Magnification 80 X. No Etch.

Oxide inclusions appear as gray areas. The black areas are pits where the oxide has dropped out during polishing of the specimen.

tion of head and web. Typical Rockwell "C" readings are shown in Fig. 6. The hardness surveys show a few spots where the diamond brale of the Rockwell-tester hit either gas pockets or inclusions on the side of the head and also some spots where this same action occurred on the horizontal slice 3/16 in. below the tread of the rail. These spots are indicated in Fig. 6 by the Rockwell "C" numbers of 3, 6 and 8. In general, the hardness numbers near the junction of the rail ends approach in magnitude the hardness numbers of the steel outside the heated zone.

A six-inch section of the rail head, including the junction between the rails, was sliced into five horizontal slices and etched in 50 percent hydrochloric acid. Figure 7 shows the under side of the top slice, which was taken about ⅛ in. below the tread of the rail. Along both sides of the head at the weld region, the acid has brought out markings which look like cracks. However, microscopic examination shows that these markings were caused by the metal having been burned during the welding operation. This action deposited iron oxide at the grain boundaries. The oxide is dissolved by the etching acid. Figure 8 is a photomicrograph of the metal in this area, showing the gray inclusions of iron oxide along the grain boundaries. Some of the oxide dropped out during the polishing of the surface of the specimen, leaving pits which show black in the figure.

IV. Mechanical Tests of Specimens From Welded Joints

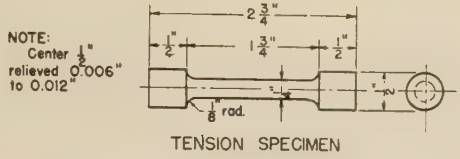
By S. W. Lyon

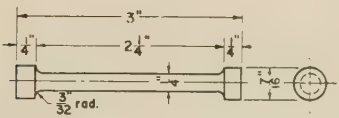
17. **Significance of Mechanical Tests of Specimens from Welds.**—In general, mechanical tests of specimens from welded-rail joints give a measure of the quality of the metal in the joint rather than a measure of the strength or toughness of the joint as a whole. In some cases, doubtless, mechanical defects were present in the small specimens cut from various parts of welded rail. But in general these tests would give an idea of the quality of the material itself when undamaged by mechanical defects—cracks, notches and other "stress raisers."

18. **Strength, Ductility and Toughness Tests of Specimens from Welds.**—The tests used to give some idea of the mechanical properties of the metal in the welds included, tension tests, fatigue tests and tension-impact tests. The tension test gives a measure of the static strength and static ductility of the material; the fatigue test gives some idea of the strength of the specimen under repeated stress; and the tension-impact test gives some measure of a combination of strength and ductility which is designated as "toughness." The specimens used are shown in Fig. 9. Tension tests were made in an Olsen screw-power testing machine using Robertson shackles to hold the specimens; fatigue tests were made in small rotating-cantilever machines which subjected the specimen to a complete reversal of flexural stress during a cycle of stress (one revolution of the machine), and tension-impact tests were made in a Charpy impact machine, fitted with a Lyon tension attachment.

Approximately 725 tensions tests, 1,280 fatigue tests, and 650 tension-impact tests have been made. Figures 11, 12 and 13 show the results of these tests.

19. **Quality of Metal in Different Parts of the Welded Joint.**—The etch tests (See Chapter III, page 746) made on different portions of the rail yielded etch patterns which show in a general way areas of section occupied by weld metal, metal affected by the heat of welding, and unaffected rail steel, as shown in Fig. 10 a. Groups of specimens were cut from the head, the web, and the base of joints, and in each





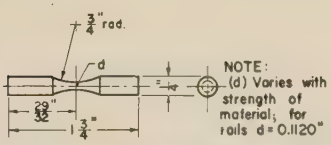
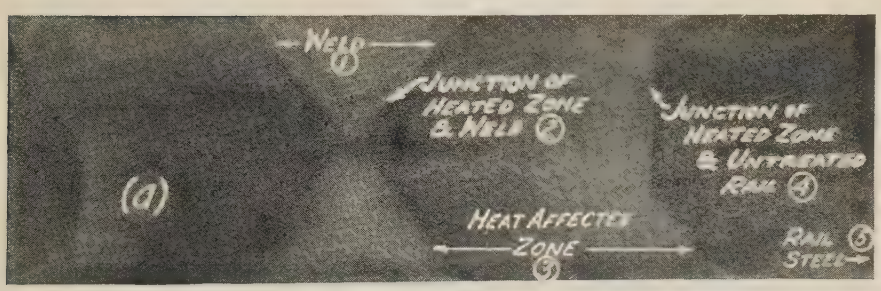


Fig. 9. Types of Specimens for Mechanical Tests of Welded Joints.



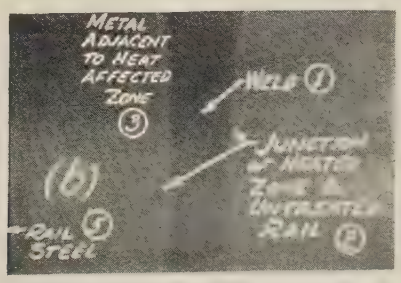


Fig. 10. Sections of Rails Welded by Processes A and D.

Etched to show extent of heated zone and location of various zones from which mechanical test specimens were machined. Approximately half natural size. Etched with Ammonium Persulphate.

(a) Gas Fusion Weld—Process A

(b) Electric Pressure Weld—Process D.

group specimens were cut from five "zones": (1) weld metal, (2) junction metal between weld metal and heat-affected metal, (3) heat-affected metal, (4) junction between heat-affected metal and rail steel, and (5) unaffected rail steel.

In this connection, as shown in Fig. 10 b, it may be noted that for the electric pressure weld (Process D) in which no metal is added during welding, and in which the heat-affected zone is very narrow, the system of zone designation as used eliminates specimens from Zone 4.

20. **Results of Mechanical Tests of Specimens.**—Figure 11 shows the results of mechanical tests of specimens cut from the rail head. In this series specimens were actually taken at each of either two or three horizontal layers in the head. The value reported is for that layer whose average value was lowest.

In the case of specimens from the web and from the base, only one layer of specimens was taken,—vertical in the web and horizontal in the base. In Fig. 11 the tensile strength seemed lowest in Zone 2, the junction metal between weld and heat-affected metal. However, for the 131-lb. rail, Processes C and D, the tensile strengths seem to be about equal to that of the rail steel, Zone 5. The endurance limit seemed to be slightly lower in the Zone 1, weld metal, and this was in general true both for the 112-lb. and 131-lb. rails, and the endurance limit of the rail steel, in general, was higher than that for the weld metal. The ductility and toughness were in general lowest in Zone 2, the junction metal between weld and heat-affected metal. Values of elongation, reduction of area, and energy for fracture (impact-tension values) were lower for weld metal than for rail steel, except for Process D, in which weld metal and rail steel showed nearly equal values for ductility and toughness.

Considering the results of specimens cut from the web, reference to Fig. 12 shows that the tensile strength in general is about equal in Zone 1, weld metal, and Zone 2, junction metal between weld metal and heat-affected metal. These values for Processes A and B are distinctly below those for rail steel. For Processes C and D the values are about the same as those for rail steel. For Process E, 131-lb. rail, the figures seem to be somewhat lower than for rail steel. The endurance limit is also lowest in Zones 1 and 2, and in all cases seems somewhat lower than for rail steel. The values for elongation, reduction of area and toughness are lowest in Zones 1 and 2, and for Processes A, B and C they are distinctly below the values for the rail steel. For Process D the results are slightly below the values for rail steel except reduction of area, which is about equal to that for rail steel. For Process E the data are incomplete, but for the 112-lb. rail the elongation and reduction of area are about equal to that of rail steel while for the 131-lb. rail and all tension-impact values the toughness is distinctly less.

Figure 13 gives a summary of the values for specimens cut from the base of the rail. The values of tensile strength again seem in general to be smallest in Zones 1 and 2. For Processes D and E the tensile strength seems to be equal to, and in some cases slightly above, that for the rail steel. For Processes A, B and C it is distinctly lower than that for the rail steel. The endurance limit is also lowest at Zones 1 and 2. In general, none of the fatigue tests show results quite up to the fatigue strength of the rail steel, although Processes D and E and the 112-lb. rail, Process C, approach that value. The ductility and toughness tests of material in the base again give minimum values for Zones 1 and 2. And again Processes D and E approach or equal the value for rail steel, and Processes A, B and C give results distinctly below. For the tension-impact results for Zones 1 and 2 it is again noted that values for Processes D and E approach (but in this case do not equal) values for rail steel, while for Processes A, B and C, values are lower than for rail steel.

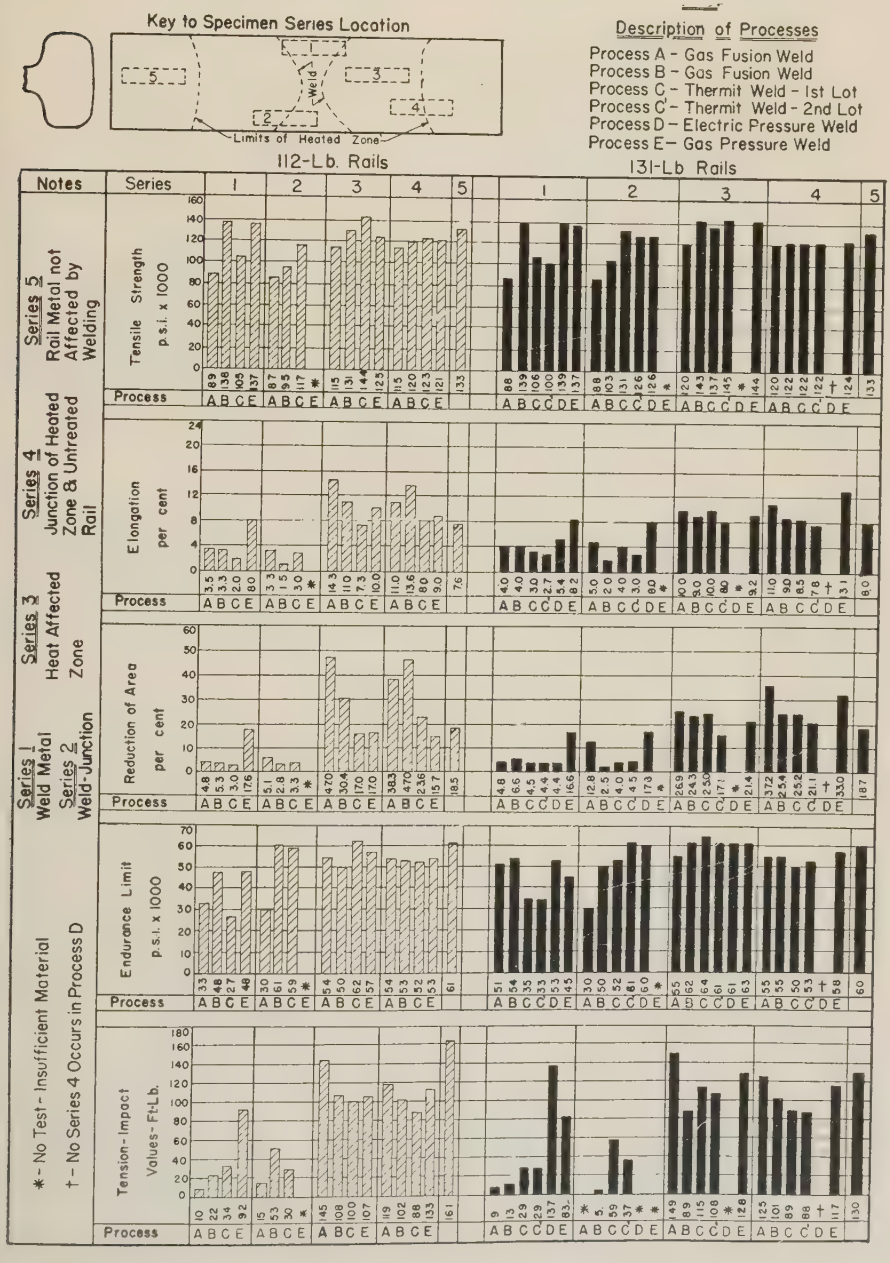

Key to Specimen Series Location
Weld
Limits of Heated Zone
Description of Processes
Process A - Gas Fusion Weld
Process B - Gas Fusion Weld
Process C - Thermit Weld - 1st Lot
Process C' - Thermit Weld - 2nd Lot
Process D - Electric Pressure Weld
Process E - Gas Pressure Weld
112-Lb. Rails
131-Lb Rails
Notes
Series
Series 1 Weld Metal
Series 2 Weld-Junction
Series 3 Heat Affected Zone
Series 4 Junction of Heated Zone & Untreated Rail
Series 5 Rail Metal not Affected by Welding
Tensile Strength p.s.i. x 1000
Elongation per cent
Reduction of Area per cent
Endurance Limit p.s.i. x 1000
Tension-Impact Values-Ft-Lb.
Process
*- No Test - Insufficient Material
†- No Series 4 Occurs in Process D

Fig. 11. Summary of Results of Mechanical Tests on Welded Rail Joints. Specimens Cut from Head.

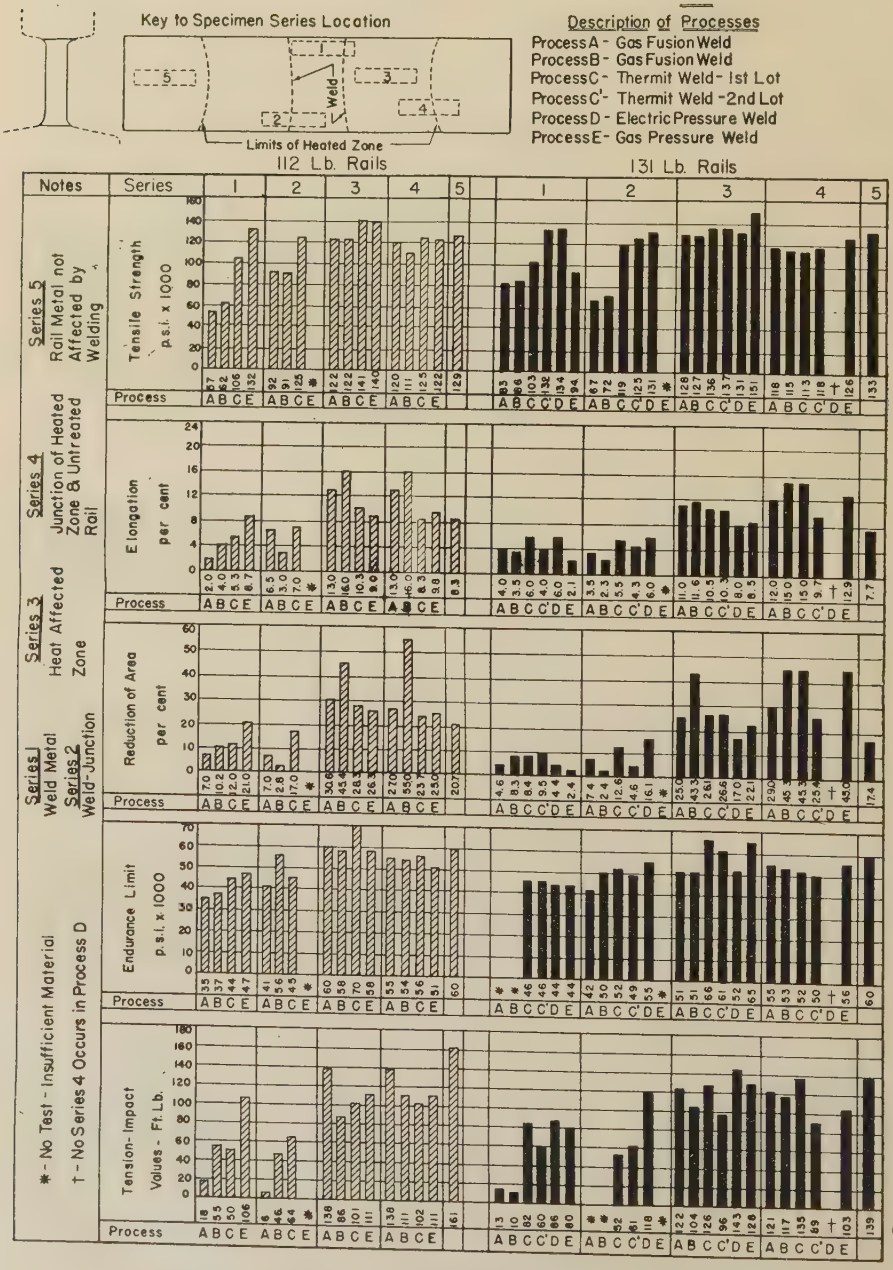

Key to Specimen Series Location
Weld
Limits of Heated Zone
Description of Processes
Process A - Gas Fusion Weld
Process B - Gas Fusion Weld
Process C - Thermit Weld - 1st Lot
Process C' - Thermit Weld - 2nd Lot
Process D - Electric Pressure Weld
Process E - Gas Pressure Weld
112 Lb. Rails
131 Lb. Rails
Notes
Series
Tensile Strength p.s.i. x 1000
Elongation per cent
Reduction of Area per cent
Endurance Limit p.s.i. x 1000
Tension-Impact Values - Ft. Lb.
Process
Series 5 Rail Metal not Affected by Welding
Series 4 Junction of Heated Zone & Unheated Rail
Series 3 Heat Affected Zone
Series 1 Weld Metal
Series 2 Weld-Junction
* - No Test - Insufficient Material
† - No Series 4 Occurs in Process D

Fig. 12. Summary of Results of Mechanical Tests on Welded Rail Joints. Specimens Cut from Web.

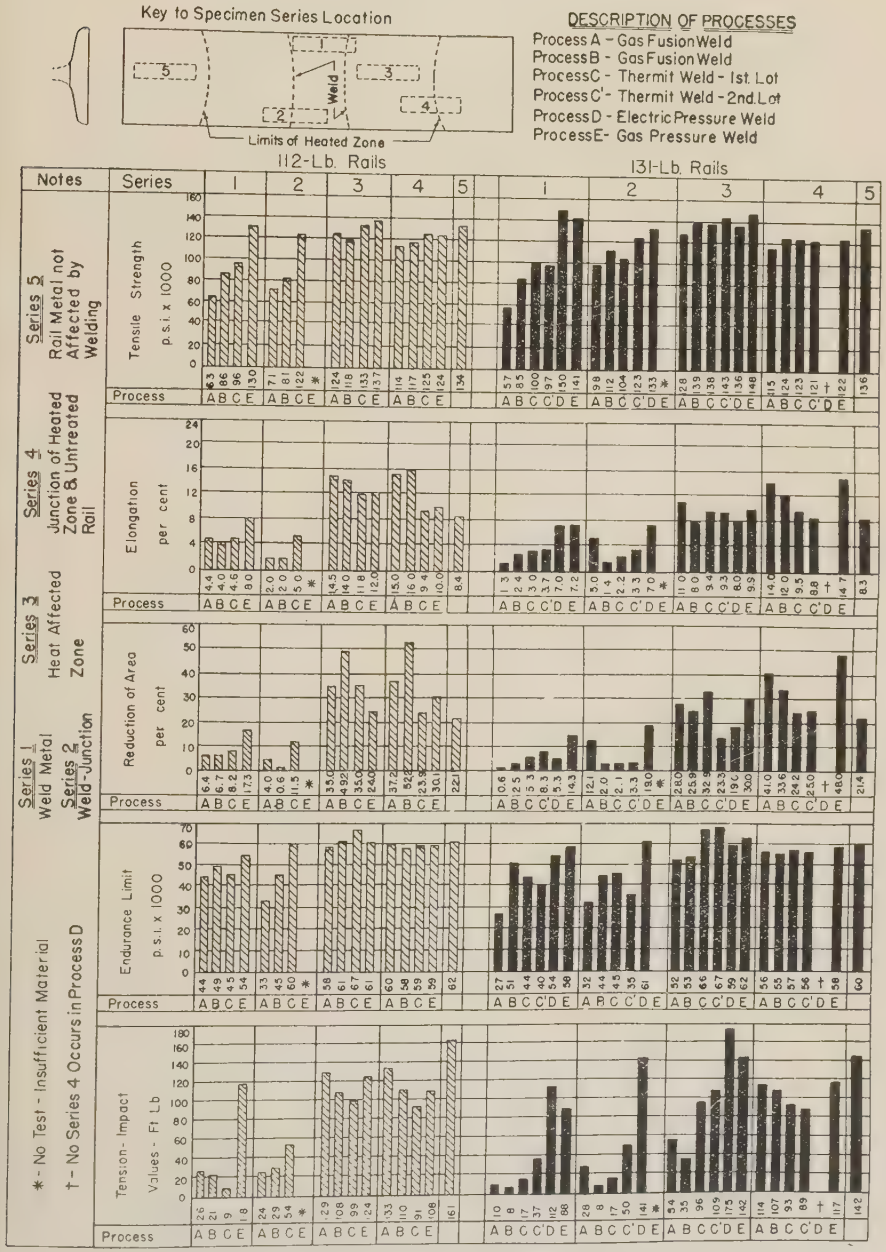

Key to Specimen Series Location
Weld
Limits of Heated Zone
DESCRIPTION OF PROCESSES
Process A - Gas Fusion Weld
Process B - Gas Fusion Weld
Process C - Thermit Weld - 1st. Lot
Process C' - Thermit Weld - 2nd. Lot
Process D - Electric Pressure Weld
Process E - Gas Pressure Weld
112-Lb. Rails
131-Lb. Rails
Notes
Series
Series 1 Weld Metal
Series 2 Weld-Junction
Series 3 Heat Affected Zone
Series 4 Junction of Heated Zone & Untreated Rail
Series 5 Rail Metal not Affected by Welding
Tensile Strength p.s.i. x 1000
Elongation per cent
Reduction of Area per cent
Endurance Limit p.s.i. x 1000
Tension - Impact Values - Ft Lb
Process
* - No Test - Insufficient Material
† - No Series 4 Occurs in Process D

Fig. 13. Summary of Results of Mechanical Tests on Welded Rail Joints. Specimens Cut from Base.

21. **Specimen Test Results as Indexes of Quality of Joints.**—As pointed out previously, the specimens cut from welded joints indicate qualities of metal rather than of the welded joints. Such factors as irregularities of outline at welds, minute cracks, and other mechanical "stress raisers" would not play much part in the tests of small specimens cut from the joint, machined to shape, and given a smooth finish. However, a comparison of the results of specimens tested with the results of rolling-load tests of joints do show various signs of the effect of poor metal on the strength of the joint as a whole. Evidently mechanical tests of specimens cut from the joint furnish one source of information concerning the strength of the joint. However, this information must be amplified by appropriate tests of the joint as a whole,—and, as in the First Progress Report, we may note that service records will furnish the final criterion.

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English Engineering Units and Their Dimensions

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A system of units based on force pounds, mass pounds, feet, and seconds is widely used in several branches of engineering, including chemical engineering. This is a four-unit system and requires the occasional use of the dimensional constant β which has the same numerical value as the standard acceleration of gravity, 32.1740, but the units mass pounds $\times$ feet/force pounds $\times$ second^2, which are not those of acceleration. The use of this dimensional constant is discussed, and examples are worked out. A clear distinction between force pounds and mass pounds is made. The application of dimensional analysis to problems in fluid motion, involving quantities evaluated in a four-dimension system, yields results which are also applicable to three-dimension systems, and a procedure is illustrated whereby the dimensional conversion factor ($\beta = Ma/F$) is automatically retained in the final result wherever needed.

THREE different consistent systems of English units are in use, in each of which length is expressed in feet and time in seconds. In the absolute system, force is expressed in poundals and mass in pounds. In the gravitational system, force is in pounds and mass in slugs. In the engineering system, which is widely used in chemical engineering, force is expressed in force pounds and mass in mass pounds. Some writers develop quantitative relations which are correct only for one or two of these systems, without showing what modifications would be made to permit the use of any of the three systems. Others present relations which require

a mixture of units taken from several of these systems, and although the correct results will be obtained by using the specified constants, the reader may be misled as to the reasons for the introduction of a certain constant. A few express the relations in nonconsistent systems, which seems inadvisable except in dimensional equations for special cases. When using engineering units involving both pounds mass and pounds force, confusion exists as to when the numerical value associated with gravitational acceleration should be included. The purpose of this paper is to clarify these problems and to indicate the relation between the three systems, without advocating exclusive use of any one. The author feels strongly that writers using the engineering system should develop general relations which will give correct results in all three systems. This can be done by including, where required, the necessary conversion factor, which equals the product of mass and acceleration divided by force.

Concept of Dimensions and Units

Engineers measure quantities of different kinds and use the relations between these measurements in engineering applications. The number of kinds of quantities which might be measured is very large. Some of the more common kinds used in mechanics are listed in Table I, column 1. An amount of any quantity is measured by comparison with a unit amount of this same quantity. The amount measured is indicated as equivalent to so many units. Fortunately it is not necessary to provide primary units for each quantity since these may be obtained in terms of primary unit amounts of some of the other quantities. Thus a volume may be measured by comparing it with a unit of its own kind (the gallon) or with a combination of units of another kind (the foot). In the first case the volume is expressed in gallons, and in the latter in cubic feet.

To describe the various quantities appearing in a given list, a minimum number of them may be selected as primary units, since the remaining (secondary) quantities may be defined in terms of these primary units. In order to obtain the minimum number of primary quantities, one must select those which cannot be measured in terms of one another and are therefore independent. More than the minimum number may be selected as primary units, but then those in excess of the minimum number can be measured in terms of the minimum number and are therefore not independent units. The kinds of quantities for which primary units are chosen, whether minimum or not, are the "dimensions", while the actual amounts of these quantities chosen as a unit are the "primary units". If length is chosen as one of the primary quantities, then length is a dimension and is here denoted by symbol L. Volume then becomes a secondary unit, involving L^3. In consistent systems of units, which are the only kind discussed in this paper, the unit of volume equals the cube of the unit of length.

Considerable latitude is possible in the choice of the primary quantities or dimensions. Thus if area, A, were chosen as a primary quantity, the dimensions of volume would be $A^{3/2}$ and of length, $A^{1/2}$. This is usually not convenient.

Only the kinds of quantities used in mechanics will be considered. However, the applications of these principles to other fields should be evident. Experience has shown that the minimum number of dimensions suitable for making measurements in mechanics is three. Length, L, and time, Θ, are commonly chosen as two of the three, and either mass, M, or force, F, as the third. All four are often chosen as primary, and then either of the last two may be expressed in terms of the other three.

The fundamental law of motion giving the relation between the quantities force, mass, and acceleration is

$$\text{force} = (1/\beta) \times \text{mass} \times \text{acceleration} \qquad (1)$$

where $1/\beta$ is a proportionality constant (*7*, *12*). With length and time as two of the dimensions, three possible choices for the others will be considered. In each, acceleration has the dimensions $L\Theta^{-2}$.

Table I. Quantities and Their Dimensions in Three Systems

Primary dimensions Name of system	M, L, Θ Absolute or dynamical or physical	F, L, Θ Gravitational or technical	F, M, L, Θ None or $FML\Theta$
Length, L	L	L	L
Time	Θ	Θ	Θ
Mass	M	$F\Theta^2L^{-1}$	M
Force	$ML\Theta^{-2}$	F	F
β, a constant	None	None	$ML\Theta^{-2}F^{-1}$
Acceleration, a	$L\Theta^{-2}$	$L\Theta^{-2}$	$L\Theta^{-2}$
Energy, work	$ML^2\Theta^{-2}$	FL	FL
Energy/unit mass	$L^2\Theta^{-2}$	$L^2\Theta^{-2}$	FLM^{-1}
Force/unit mass, a/β	$L\Theta^{-2}$	$L\Theta^{-2}$	FM^{-1}
Local acceleration of gravity, g_L	$L\Theta^{-2}$	$L\Theta^{-2}$	$L\Theta^{-2}$
Diameter, D	L	L	L
Pressure, p	$ML^{-1}\Theta^{-2}$	FL^{-2}	FL^{-2}
Power	$ML^2\Theta^{-3}$	$FL\Theta^{-1}$	$FL\Theta^{-1}$
Mass velocity, G	$M\Theta^{-1}L^{-2}$	$F\Theta L^{-2}$	$M\Theta^{-1}L^{-2}$
Linear velocity, V	$L\Theta^{-1}$	$L\Theta^{-1}$	$L\Theta^{-1}$
Momentum	$ML\Theta^{-1}$	$F\Theta$	$ML\Theta^{-1}$
Absolute viscosity, μ	....	$F\Theta L^{-2}$	$F\Theta L^{-2}$
Absolute viscosity, $\mu' = \mu\beta$	$ML^{-1}\Theta^{-1}$	....	$ML^{-1}\Theta^{-1}$
Density, ρ	ML^{-3}	$F\Theta^2L^{-4}$	ML^{-3}
$DV\rho/\mu$	.....	None	$MLF^{-1}\Theta^{-2}$
$DV\rho/\mu' = DV\rho/\beta\mu$	None	....	None
$\Delta p/\rho V^2$	None	None	$F\Theta^2M^{-1}L^{-1}$
$\Delta p\beta/\rho V^2$	None	None	None
Surface tension, σ	$M\Theta^{-2}$	FL^{-1}	FL^{-1}

The absolute or dynamical (*5*) dimension system selects mass as the third dimension. A quantity of force is measured as the product of a number of units of mass and a number of secondary units of acceleration. Thus in this system of dimensions, by definition force = mass × acceleration, and force has the dimensions $ML\Theta^{-2}$. Equation 1 indicates that

$$[ML\Theta^{-2}] = \frac{1}{\beta}[M][L\Theta^{-2}] \qquad (2)$$

and therefore β is unity and has no dimensions.

The gravitational or technical dimension system selects force as the third dimension and measures a quantity of mass as a number of units of force divided by a number of units of acceleration. Then by definition, mass = force/acceleration and the dimensions of mass are $FL^{-1}\Theta^2$. Equation 1 gives

$$[F] = \frac{1}{\beta}[FL^{-1}\Theta^2][L\Theta^{-2}] \tag{3}$$

and again β is unity and has no dimensions.

A four-dimension system uses force, mass, length, and time as primary (*2*). This system has been used for a number of years in many fields, such as engineering thermodynamics and chemical engineering. It will be referred to as the $FML\Theta$ dimension system. Here a quantity of mass is measured in primary units of mass and a quantity of force in primary units of force. The latter unit is not equal to the product of unit mass and unit acceleration but is defined independently. Equation 1 now gives

$$[F] = \frac{1}{\beta}[M][L\Theta^{-2}] \tag{4}$$

and β has the dimensions $ML\Theta^{-2}F^{-1}$ and a numerical value determined by the relative sizes of the four primary units.

Table I gives a list of quantities and their dimensions in the three systems discussed. The dimensions in column 4 for the $FML\Theta$ system are so chosen that force has the dimension F and mass, M, except in the case of viscosity which is given in terms of both $FL\Theta$ and $ML\Theta$, distinguished by the symbols μ and μ', respectively. (The decision as to whether a given kind of quantity involves force or mass in its original concept is not always possible. Thus viscosity may be defined as force per unit area per unit velocity gradient which indicates the dimensions $F\Theta L^{-2}$, or it may be defined, as in the kinetic theory, as the rate of momentum transfer per unit area per unit velocity gradient which would make its dimensions $ML^{-1}\Theta^{-1}$.) Within the $FML\Theta$ system the dimensions of a quantity expressed in F, L, and Θ may be changed to M, L, and Θ by multiplying each F dimension by the dimensions of β, $ML\Theta^{-2}F^{-1}$. The numerical value of the quantity may be converted by multiplying by the numerical value of β for the particular unit system used.

Definition of Units

The actual size of the primary units has not been mentioned, and the above discussion is therefore independent of the amount chosen for each primary unit. Some of the common English units for force and mass will be defined. The units for length and time are taken as feet and seconds, and their definitions do not need to be repeated here.

The English absolute or dynamical unit system uses as primary unit mass the avoirdupois pound or mass pound,

which is defined in this country in terms of the United States prototype kilogram 20 as 0.4535924277 kg. (*9*). The unit of force is a secondary unit, the poundal, and is the force required to give one mass pound unit acceleration or

$$1 \text{ poundal} = \frac{1 \text{ mass pound}}{1} \times 1 \text{ ft./sec.}^2 \qquad (5)$$

This definition makes the force of gravity on a mass pound numerically g_L poundals of force where the local acceleration of gravity is g_L feet/second^2.

The English gravitational unit system uses as a primary unit of force the standard force pound, which is defined as the force which will give the avoirdupois pound an acceleration of g_0 feet/second^2, where g_0 is the internationally adopted standard acceleration of gravity, 980.665 cm./second^2 or, to the fourth decimal place, 32.1740 feet/second^2. The unit of mass is a secondary unit, the slug, and is the amount of mass which will be accelerated 1 foot/second^2 when acted upon by one standard force pound or

$$1 \times 1 \frac{\text{standard force pound}}{1 \text{ ft./sec.}^2} = 1 \text{ slug} \qquad (6)$$

This makes the slug equal to 32.1740 mass pounds.

The term "force pound" defined above and used hereafter is the standard gravitational force pound. This should be distinguished from the gravitational force pound (*5*) which is defined so that the local force of gravity acts on one avoirdupois pound (one mass pound) with one gravitational force pound. This latter definition results in a force unit which is not a definite fixed quantity of force but varies with g_L from place to place so that the force of gravity acting on a body when measured in gravitational force pounds is always numerically equal to its mass in mass pounds. All scales or balances are calibrated (*4*) locally against standard "weights" (standards of mass). (The weight of a body is variously interpreted as the mass of the body and also as the force of gravity acting on the body. It is therefore necessary to state definitely the units used in expressing a weight. Standard "weights" are, however, standards of mass, *9*.) Then, the reading or weight is obtained in either mass-pound or gravitational force-pound units. The latter may be converted to standard force pounds by multiplying by the ratio g_L/g_0 (*8*) or

$$\text{local force of gravity (standard force pounds)} = \frac{\text{weight (gravitational force pounds)}}{g_0} \times g_L \qquad (7)$$

As shown later the weight in mass pound units may be converted to standard force pounds by multiplying by g_L/β, or

$$\text{local force of gravity (standard force pounds)} = \text{weight} \frac{(\text{mass pounds}) \times g_L}{\beta}$$

Another concept common among engineers (*11*) is that mass is simply a ratio of weight (i. e., the force of gravity in standard force pounds) to g_L. Although the secondary unit of mass is not given a name in this case, this ratio is obviously the same as the mass measured in slugs.

TABLE II. ENGLISH UNITS

Name of System	Absolute	Gravitational	Engineering
Mass	Mass pound	Slug (or gee pound)	Mass pound
Force	Poundal	Force pound	Force pound
Length	Ft.	Ft.	Ft.
Time	Sec.	Sec.	Sec.
β	Unity	Unity	$32.1740 \frac{\text{mass lb.} \times \text{ft.}}{\text{force lb.} \times \text{sec.}^2}$

Relative Size of Units

$$1 \text{ mass lb.} = \frac{1 \text{ force lb.} \times 1 \text{ sec.}^2}{32.1740 \times 1 \text{ ft.}} = \frac{1 \text{ slug (or gee lb.)}}{32.1740} = \frac{1 \text{ poundal} \times 1 \text{ sec.}^2}{1 \text{ ft.}}$$

$$1 \text{ force lb.} = \frac{32.1740 \times 1 \text{ mass lb.} \times 1 \text{ ft.}}{1 \text{ sec.}^2} = \frac{1 \text{ slug (or gee lb.)} \times 1 \text{ ft.}}{1 \text{ sec.}^2} = 32.1740 \text{ poundals}$$

The English engineering units use as a primary unit of force, the force pound, and as a primary unit of mass, the mass pound defined above. The relation between measurements in the four primary units is given by Equation 1:

$$\text{force pounds} = (1/\beta) \times \text{mass pounds} \times (\text{ft./sec.}^2) \qquad (8)$$

β, therefore, has the units (mass pounds $\times$ feet)/(force pounds $\times$ second^2); and from the definition of a force pound as the force which will give one mass pound an acceleration of 32.1740 feet/second^2, the numerical value of β is 32.1740. β is a dimensional constant and has nothing to do with the acceleration of gravity, either local or standard, except that its numerical value happens to be equal to that of the standard acceleration of gravity, g_0. This is in contradiction to numerous cases in the literature where the equivalent of β is stated to be the acceleration of gravity.

Table II shows several sets of English units. In the absolute and gravitational unit systems, the poundal and the slug are secondary units; all the other units are primary. Table II also gives the relative size of the units which may be used to measure quantities of mass and force.

Examples of Use of Dimensions

Dimensionless equations are valid, regardless of the size of the consistent units chosen for each dimension. This feature can be used to predict how the kinds of quantities associated with any operation or process must be grouped in an equation and when so used is called "dimensional analysis". The methods of dimensional analysis are adequately discussed by Bridgman (*2*), and numerous examples are given in the literature of these methods applied to the systems of three dimensions. Since chemical engineers use the four-dimension system, their dimensional analyses should be carried out in

terms of four dimensions. The following example will clarify this point.

Example 1. Consider the case of Poiseuille's law for the friction losses during streamline flow through a cylindrical tube, and suppose, for instance, that the relation between the variables (quantities) is not known and is sought by dimensional analysis. The variables are $-dp$, μ (or μ'), dL, V, and D as defined in Table I. If the $ML\Theta$ system is chosen, viscosity μ' is used; and since β is dimensionless, it need not be considered. The above is therefore a complete list of the quantities involved. If the $FL\Theta$ system is chosen, viscosity μ is used, and again β is dimensionless and no other terms need be considered. If the $FML\Theta$ system is used, the viscosity can be either μ or μ The quantity μ' will be chosen since this is customary for chemical engineers. Now β has the dimensions $ML\Theta^{-2}F^{-1}$, and it may or may not be necessary to add it to the list of variables. This point will be discussed.

Consider the statement that in the final equation the dimensions of each term must be the same and apply this requirement to the dimension force, F. If one of the dimensions of dp is chosen as F, then this statement may be satisfied by making F one of the dimensions of viscosity since none of the other variables can be measured in terms of F. Now $-dp$ can appear on one side of the equation and μ on the other, and the requirement will be satisfied. However, it has been agreed to use μ' in this example, which makes M, not F, one of the dimensions of viscosity, and there is only one F and one M in the list of variables. There is now no way to satisfy the statement except by including β, which is not a variable but does contain both F and M in its dimensions. Therefore, when the list of variables known to be involved in a problem contains both the dimensions F and M, but when either is included only once, the dimensional constant β should be added to the list. Otherwise it may be omitted.

Returning now to the relation between these six quantities in the Poiseuille law problem and using brackets to represent dimensions but not numerical values,

$$[-dp][\mu']^a[dL]^b[V]^c[D]^d[\beta]^e = 0 \qquad (9)$$

or substituting the dimensions of each quantity,

$$[FL^{-2}][ML^{-1}\Theta^{-1}]^a[L]^b[L\Theta^{-1}]^c[L]^d[MLF^{-1}\Theta^{-2}]^e = 0 \qquad (10)$$

then $\Sigma F = 1 - e = 0$
$\Sigma M = a + e = 0$
$\Sigma L = -2 - a + b + c + d + e = 0$
$\Sigma\Theta = -a - c - 2e = 0$

and solving these equations gives $e = 1$, $a = -1$, $c = -1$, and $d = 1 - b$ or

$$[-dp][\mu']^{-1}[dL]^b[V]^{-1}[D]^{1-b}[\beta] = \left[\frac{-dpD\beta}{\mu' V}\right]\left[\frac{dL}{D}\right]^b \qquad (10A)$$

Therefore the dimensionless ratio $(dpD\beta/\mu' V)$ is a function of the ratio dL/D. The pressure drop $-dp$ is known to be

proportional to the tube length dL, and this additional fact may be used to simplify the relation to

$$-\left(\frac{dpD\beta}{\mu' V}\right)\left(\frac{D}{dL}\right) = C \qquad (11)$$

A mathematical analysis based on the equilibrium between the frictional drag and the pressure drop would give the same formula, together with the numerical value of the constant C as 32. This latter is a dimensionless constant and has the same value, regardless of the consistent units used to express the quantities in Equation 11. It is not related to the acceleration of gravity or to β. The dimensional constant β will vary with the size of the units. As shown above, β equals 32.1740 (mass pounds $\times$ feet/force pounds $\times$ second^2).

The general viewpoint can be further illustrated by choosing other units for Equation 11. If dp and μ' are expressed in poundals/square foot and pounds mass/feet $\times$ second, respectively, according to the English absolute system of units β is unity and Equation 11 becomes:

$$-\left(\frac{dpD}{\mu' V}\right)\left(\frac{D}{dL}\right) = C \qquad (12)$$

In these units the numerical value of dp will be 32.1740 times that in the first set.

In deriving Equation 11, a prior knowledge of the exact list of variables important in the problem was necessary. This knowledge must be based on experiments. Conversely, the validity of an equation derived by a dimensional analysis based on a list of variables assumed to be the important ones must be proved by comparing it with the results of experiments.

Application of Units

Table II indicates that a choice of English units is available for measuring any quantity. This choice often leads to minor differences in dimensionless equations or groups of variables. The following examples will illustrate these differences.

Example 2. Chemical engineers have used the viscosity μ' in mass pounds/foot $\times$ second, which is obtained from the relation $\mu' = 0.000672 \times$ centipoises. On the other hand, the steam tables of Keenan and Keyes (*6*) give μ as force pounds $\times$ second/square foot. According to Equation 8, $\mu\beta = \mu'$ where β is 32.1740 (mass pounds $\times$ feet/force pounds $\times$ second^2).

With the units familiar to chemical engineers as listed in column 4 of Table I, Poiseuille's law is written correctly as

$$-\left(\frac{dpD}{\mu V}\right)\left(\frac{D}{dL}\right) = 32 = -\left(\frac{dpD\beta}{\mu' V}\right)\left(\frac{D}{dL}\right) \qquad (13)$$

and β does not appear when both dp and μ are expressed in force pound units as on the left, but it must be included to make the equation dimensionless when μ' is used in mass pound units as on the right.

EXAMPLE 3. Reynolds' number written in the absolute system is $DV\rho/\mu'$ and is dimensionless. In the gravitational system it is $DV\rho/\mu$ and is dimensionless. In the $FML\Theta$ system it is dimensionless when written as $DV\rho/\mu'$ which is numerically equal to $DV\rho\beta/\mu$, also dimensionless.

EXAMPLE 4. Bernouilli's mechanical energy balance is often derived by equating the mechanical energy input to the output for each mass pound of fluid entering and leaving an apparatus. With the height above a given datum plane measured as Z, the potential energy per mass pound (so-called potential head) is given as

$$\frac{1 \text{ mass pound} \times Z \times g_L}{1 \text{ mass pound} \times \beta} = Z\frac{g_L}{\beta}$$

in the engineering unit system and has the units (force pounds $\times$ feet)/mass pounds. The ratio g_L/β is nearly unity and has formerly been omitted; this practice is confusing.

EXAMPLE 5. The kinetic energy per mass pound of fluid in the engineering unit system is

$$\frac{1 \text{ mass pound} \times V^2}{1 \text{ mass pound} \times 2 \times \beta} = \frac{V^2}{2\beta}$$

and has the units (force pounds $\times$ feet)/mass pounds.

In books on chemical engineering it has been the practice to write $V^2/2g$ for the kinetic energy term for work per unit mass, where g is said to be the acceleration due to gravity, equal to 32.2 feet/second^2. Although $V^2/2g$ gives the correct numerical result, the quotient is not force pounds times feet divided by pounds mass, whereas $V^2/2\beta$ actually gives work per unit mass.

EXAMPLE 6. In the $FML\Theta$ system, the flow-work term equals

$$pv = \frac{p}{\rho} = \frac{F}{L^2} \times \frac{L^3}{M} = \frac{FL}{M}$$

and hence the sum of the potential, kinetic, and flow-work terms is

$$\frac{Zg_L}{\beta} + \frac{V^2}{2\beta} + pv \qquad (14)$$

The present practice of writing

$$Z + \frac{V^2}{2g} + pv \qquad (14A)$$

gives the same numerical result as Equation 14 but tends to be confusing. In fact, Equation 14 is general for all three systems. In any one of the three consistent systems all three terms have the same dimensions and are properly additive, and each term is expressed as work per unit mass—namely, FL/M for the $FML\Theta$ system and L^2/Θ^2 for the $ML\Theta$ and $FL\Theta$ systems.

Upon examining a number of problems in fluid motion, with the pertinent factors evaluated in the $FML\Theta$ system, it is found that β or g_L alone, or both, or neither may appear

in the final dimensionless equation. Where the conversion factor does appear, the result is also applicable, in the $ML\Theta$ and $FL\Theta$ systems. In the latter two cases, where β equals 1 and is dimensionless, it disappears. Because of this, readers familiar with systems where β equals 1 are apt to become confused when working with the $FML\Theta$ system. It was shown above that, when working with the $FML\Theta$ system, a result applicable to all three systems is obtained. Having done this the only precaution one need take is to evaluate each dimensionless group from one of the three consistent systems.

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NEW TRENDS IN BOILER FEED WATER TREATMENT*

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This paper was prepared for presentation at The Berlin Congress scheduled for June, 1940. Upon cancellation of the Congress, this paper was released to the Institute and appeared on the program of the Buffalo-Niagara Falls, N. Y., meeting, May 13-15, 1940.

Summary

The paper reviews the new trends in boiler water treatment for the prevention of scale, corrosion, embrittlement and carry over. These include the latter developments in cold process softeners and the organolites or Zeo-Karb softeners. Consideration is given to the latest methods used for prevention of silica scale, as well as the so-called "Threshold treatments" for stopping scale in condensers and heat exchangers. The newer types of steam purifiers are described. There is also a discussion on corrosion in the "steam blanketed" areas of the boilers.

"If any apology is due for discussing the somewhat perennial subject of boiler water, it might be fair to say, perhaps, that the matter is coming to be treated in a more rational manner; that much of the quackery and mysticism that has so long prevailed is being superseded by a more scientific and sensible consideration of the problem." [9] During the intervening period of more than four decades, since this was penned, a continuation of the "scientific and sensible consideration of the problem" has enabled the chemical engineers to keep pace with the various water treating problems which have presented themselves.

The advent of higher steam pressures, increased rates of heat transfer, and decrease in the number of steaming units in the power house has placed increased responsibility for proper boiler feed water treatment on the chemical engineer. Partridge and Purdy [11] have very ably described the problems confronting the chemical engineer

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in the modern power plants. The most common classification of these problems is as follows:

1. Scale
2. Corrosion
3. Embrittlement
4. Carry Over

The first two difficulties may be experienced at almost all points in the water-steam cycle of the power plant whereas the last two occur only in the steam boiler. There are many methods of boiler water treatment which prove suitable for one or all of these problems. However, the methods of attack may be roughly classified as treatments used internal or external to the boiler. In the more recent years it has proved advisable to use a combination of both the external and internal treatments.

Scale

The external softening of water for the removal of scale forming materials such as calcium, magnesium and silica has received increased attention with marked results in the last few years. The recirculation of sludge [5,14] with increased rate of softening in the cold process lime softener has resulted in the wider application of the cold process softener as a primary treatment ahead of zeolite and hot process softeners. This change in reaction rate has been accomplished by taking advantage of the increased rate of chemical reaction which results when the sludge resulting from the precipitation within the softener is thoroughly mixed with the treating chemicals and the water being treated. The sludge is agitated by means of revolving paddles or blades near the bottom of the softener. This brings the treating chemicals, the sludge and the incoming water all in intimate contact. The water then flows up at diminishing velocity through a floating sludge blanket. This is accomplished by the design of the softener compartments. The water leaves the sludge blanket almost crystal clear and the chemical reaction is complete. Thus in a comparatively short space of time a clear, soft stable water has been produced in the cold with a minimum of chemicals. As a precaution, a filter is usually used after these softeners; however, the time interval between washing the filters is greatly lengthened. Very little if any precipitate is formed on the filter beds. The application of proper coagulants and

control of pH value in this method has also resulted in marked reduction of the silica content of some waters being treated.[13,8]

The modification of the zeolite method of softening so as to be able to use the hydrogen as well as sodium cycles has given a new method of eliminating the positive and in some cases the negative ions also from the water.

The older zeolite softener involved the use of the siliceous zeolite containing sodium aluminum silicates. These were not stable in acid or strongly alkaline waters. This made it necessary for close control of the pH value of a water being treated in the older base exchange softeners. A new softening material for use in the base exchange softeners now allows its use in acid or alkaline waters. These bodies are called, "Organolites," "Zeo-Karb," "Exchange Filters," "Demineralization," etc. They operate under two conditions, either on the sodium or the hydrogen cycle. They may be used in a manner similar to the older zeolites and be regenerated with a salt or brine solution. Under such conditions they are said to be operating on the sodium cycle and they give a treated water having the same composition as one treated with the older type zeolites. They do, however, have the advantage of being non-siliceous and thus preclude the possibility of increasing the silica content of the water. They are also more stable over wider range of pH changes than the older zeolites. Under the conditions of the sodium cycle they convert the calcium and magnesium bicarbonates to sodium bicarbonates and the sulfates or chlorides to sodium sulfate or chloride.

If they are regenerated with an acid solution such as sulfuric acid instead of a salt solution, they are said to be operating on the hydrogen cycle. After they have been treated with acid they give an effluent in which the calcium and magnesium bicarbonates are converted to carbonic acid (H_2CO_3) and the chlorides and sulfates to hydrochloric and sulfuric acid respectively. The carbonic acid readily breaks down to form carbon dioxide and water. The carbon dioxide is readily removed by blowing air through the water (degasification). The effluent from the hydrogen cycle is acid in nature and could not readily be used for boiler feedwater. The usual procedure followed is to pass part of the raw water through one tank operating on the sodium cycle, which gives an alkaline water containing sodium bicarbonate. The rest of the water is passed through the hydrogen cycle and the effluent from the two softeners mixed in the proper proportions so as to give the desired alkalinity. The mixed water is then degasified.

The equipment used in the acid cycle must be properly protected so as to prevent acid attack on the metal parts.

The water resulting from the use of a combination sodium and hydrogen cycle will have a reduction in total solids depending on the relation between the bicarbonates and the chlorides and sulfates in the raw water. As the bicarbonates increase, there is more reduction in total solids. At the same time the cost of treatment increases since more sulfuric acid is used. This treatment is in general more expensive than the older zeolite process. The equipment cost is higher since acid-resisting coatings must be used and the newer softening materials are more expensive. The increased cost, however, is more than offset in many installations by the reduction in total solids with resulting decrease in blowdown and decrease in makeup water.

These new base-exchange softening materials are low in silica whereas the older type of zeolites had a high silica content. The use of these materials thus prevents the contamination of the treated waters with silica. The hydrogen cycle has proven quite effective in removing ammonium compounds from the water.

Of course, none of these methods remove all the hardness, and additional chemical treatment is often essential to prevent scale formation. The most common method in use has been the use of phosphates for prevention of sulfate or carbonate scale. Silica scale has been encountered in boilers having areas of high heat transfer and fed with waters containing appreciable silica. The method of attack for silica scale prevention being emphasized at the present time is the removal of silica from the water prior to entering the cycle. This has involved the use of coagulants such as iron, aluminum, and magnesium salts, which under proper pH control prove quite effective in lowering the silica content.[8,13] In general these methods have proven to be somewhat expensive. The use of the nonsilicious zeolites and anthracite coal filters serve to prevent the pick-up of silica which was so common years ago. In some installations the internal use of finely divided iron and iron or zinc salts [19] has aided in retarding silica scale formation. However, the use of any material which causes a precipitate or deposition within the boiler, whether phosphate, iron, or zinc has quite often resulted in sludge deposition in areas of sluggish circulation with resultant loss of tubes. Consequently the trend has been toward the most efficient external removal of scale or sludge forming material with the minimum of internal treatment for the higher-pressure higher-rated plants. It has been found

that the silica deposits form only in areas of high heat transfer and by changing the operating conditions so as to eliminate these spots of excessive heat transfer the scale has been prevented.

One type of scale encountered in many plants has been deposits in feed lines, turbine condensers, heat exchangers, etc. This has resulted in decrease in operating efficiency as well as increase in maintenance cost. In the past this has been prevented by treating with small amounts of organic materials. Recent studies have shown [12] the effectiveness of very small amounts of inorganic chemicals such as molecularly dehydrated phosphates (meta-, pyro-, and tetra-phosphates). This use of what has been termed, "threshold treatment," has shown that a few parts per million of this type of salt will prove quite effective in preventing scale in turbine condensers, heat exchangers, etc.

Corrosion

The corrosion most commonly experienced in the power plant has resulted from dissolved oxygen. The modern plants have resorted to every method available for the removal of the oxygen from the water prior to entering the boiler feed lines. This has resulted in almost universal chemical treatment being resorted to for the elimination of the last trace of oxygen left after deaeration. Chemicals such as iron, ferrous hydroxide, sodium sulphite [7,17] and various organic materials are being extensively used.

In areas which are "partially dry" or "steam blanketed" corrosion occurs at an accelerated rate.[6,16] The only safe method of elimination of this type of corrosion has been the elimination of the dry areas by changing circulation or points of high heat imput with the subsequent wetting of the complete tube area.

Embrittlement

Embrittlement in steam boilers results in the failure of the steel in the riveted areas and the areas in the vicinity of the rolled tube ends. This failure has been attributed to the action of the sodium hydroxide in the boiler water concentrating in the capillary spaces present in these areas and attacking the highly stressed boiler metal with the resultant cracking of the steel. Methods of chemical treatment have been worked out whereby the embrittling action of the water may be stopped. This has involved the maintenance of definite

amounts of sulfates and chlorides or organic [10,15] material in the lower-pressure boilers and the reducing of the silica contents in the higher-pressure boilers. Naturally the reducing of the sodium hydroxide content of the boiler water will tend to reduce the tendency toward embrittlement at all pressures. In the newer welded drums the riveted areas have been eliminated and the tube ends are the only remaining areas of potential danger. The methods of rolling the tube ends so as to keep the tube in contact with the drum or header surface on the inside tends to reduce the potential possibility of concentration of the boiler water in these areas.

A recent survey made in the United States indicates that failure of this type has not been experienced in boilers operating at pressures above 600 lbs. irrespective of water treatment. However, all of these boilers had either inside calked seams, or welded or forged construction.

In the lower-pressure boilers no cases of embrittlement have been encountered in boilers where the recommended A.S.M.E. (sodium sulfate to total alkalinity) ratios have been maintained. Many instances of embrittlement have occurred when these ratios have not been maintained. Consequently the boiler code recommendations may be said to have aided materially in preventing this type of trouble.

Carry Over

The presence of small amounts of solids in the steam quite often results in difficulty in the form of turbine-blade deposits. In order to reduce the total solids in the steam without reducing those in the boiler water, methods of steam cleaning or purification have been developed. One method in general makes use of the principle of washing or scrubbing the steam with the water being fed to the boiler. If this water enters the boilers at the temperature of the boiler water and is brought in intimate contact with the steam, the boiler water in the steam will be replaced with feed water. Since the feed waters usually have a total solids content less than one-tenth that of the boiler water, the total solids in the resulting steam will be reduced to a low figure without changing those in the boiler water. In addition to reducing the total solids, the composition of the solids is also changed. Thus a boiler water containing sodium hydroxide might give a steam having an appreciable sodium hydroxide content, but after being washed with a feed water having a low

sodium hydroxide content the sodium hydroxide content of the steam might be eliminated.

In some recent installations it has been possible to remove sufficient entrained boiler water to reduce or eliminate the steam washing.[2] This has involved the use of a large number of so-called cyclone separators within the boiler drum. In these separators the boiler water is mechanically separated from the steam so that the steam contains an extremely small amount of entrained boiler water.

Recent experience indicates the necessity for further study relative to the effect of impurities in the steam on the endurance limit of the metals used for the steam turbine blades. It may prove necessary to adopt types of blade materials which are non-reactive with the impurities commonly encountered in boiler waters.

The results which have been obtained in the newer steam power plants show that there has been a close cooperation between the chemical engineers, the designing engineers, and the operators. This cooperation has started from the early design of the plant and extends through its ultimate operation. This has resulted in the prevention of many difficulties and aided in obtaining increased efficiency in the steam power plant.

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